JOURNAL

OF THE

AMERICAN SOCIETY OF NAVAL ENGINEERS

VOL. XXIII. MAY, 1911. No. 2.

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DESCRIPTION AND TRIALS U. S. TORPEDO BOAT DESTROYERS *McCALL, BURROWS* AND *AMMEN.*

BY LIEUTENANT C. W. DENSMORE, U. S. N., MEMBER.

Torpedo Boat Destroyers No. 28, the *McCall,* and No. 29, the *Burrows,* were authorized by Act of Congress May 15, 1908. Contracts with the New York Shipbuilding Company, Camden, N. J., for their construction were signed October 5, 1908. The vessels were to be completed and ready for delivery by October 5, 1910.

The contract price for each vessel was $665,000.00, of which $390,000.00 was allotted to the machinery.

The keel of the *McCall* was laid June 8, 1909, and the vessel launched June 4, 1910.

The keel of the *Burrows* was laid June 19, 1909, and the vessel launched June 23, 1910.

The official contract trials of the *McCall* were begun September 6, 1910, but were not completed on account of necessary alterations to the fuel-oil burners. A second trial was carried out successfully December 8–12, 1910, and the vessel

was completed and delivered at the Navy Yard, Philadelphia, Pa., January 18, 1911.

The *Burrows* was given her official contract trials January 7–10, 1911, completed, and delivered at the Navy Yard, Philadelphia, Pa., February 17, 1911.

The general requirements of the contracts were as follows: Trial displacement about 742 tons; speed on four-hour full-speed trial not less than 29½ knots.

Torpedo Boat Destroyer No. 35, the *Ammen*, is one of the six vessels of this class authorized by Act of Congress of March 3, 1909.

Contract with the New York Shipbuilding Co., was signed June 18, 1909. The keel was laid March 30, 1910, and the vessel launched September 20, 1910.

The contract price was $648,000.00, of which $378,000.00 was allotted for machinery.

GENERAL DESCRIPTION OF THE VESSELS.

The three vessels are in all essential particulars alike. The hulls are of steel, galvanized in all parts below the water line, and lightened by omitting or cutting out all useless material.

There is a main deck and raised forecastle, the latter extending as far aft as frame 41. On the forecastle deck are located the anchors and anchor davits, capstan and pilot house, with bridge above. Under the forecastle is a small storeroom in the eyes, lamp room and windless-engine room extending to frame 12, where a watertight bulkhead is worked. Abaft of this bulkhead are quarters for four commissioned officers extending aft as far as frame 34. From frame 34 to frame 41 a passageway is formed on the starboard side. The ship's galley is amidships, with the crew's washroom and water closet on the port side. Abaft the galley, on the center line, is a companion way leading down to the crew's space.

There are two pole masts, four smoke pipes, true ellipses in section, raised hatch coamings for the firerooms, four forced-draft down-takes with mushroom hoods that can be

raised to form deflectors for the wind or battened down to keep out the sea or weather. The air ducts are about waist high when closed.

Abaft the smoke pipes on deck is the engine-room hatch, abaft of which is a deck house, including a crew's water closet and wash room, wireless telegraph room, and hatchway to the berth deck.

Abaft the deck house, at about frame 149, is located the hand-steering gear.

The berth deck extends as far aft as frame 51, and from frame 124 to frame 160. On the berth deck a small store room is located in the fore peak, abaft which is the paint locker.

Between frames 12 and 51 are the petty officers' quarters and crew space, separated by a watertight bulkhead at frame 28, provided with a watertight door; the petty officers' space being forward of the crew space. The only access to the petty officers' space is through the crew space. There are also bulkhead ladders at frame 51, port and starboard, giving access from the crew space to the main deck. The forward storerooms, on both the main and berth decks, are reached from the forecastle. The living spaces are all sheathed with No. 27 U. S. G. galvanized iron, cork painted. Sixteen metal folding berths and one mess table are provided in the petty officers' quarters, and forty-four berths and two mess tables in the crew's space. These quarters and store rooms are lighted by dead lights and air ports in the ship's side. There is no artificial ventilation excepting electric fans in the living spaces.

The after berth deck is divided as follows: From frame 124 to frame 138 are the quarters for the men of the engineer's force. From frame 138 to frame 149 are quarters for chief petty officers. Eighteen metal folding berths and one mess table are fitted in the former and eight berths and one mess table in the latter compartment. Abaft the chief petty officers' quarters are ordnance, engineer's and equipment store rooms, and sail room. The living spaces and store rooms on the berth deck, aft, are provided with air ports and dead lights, the same as the forward compartments.

There are two boiler compartments and one engine compartment. The boiler compartments, extending from frame 59 to frame 97, are provided with fuel-oil settling tanks, outboard from the boilers, their tops forming a shelf on which are located pumps, filters, heaters, etc. The engine compartment extends from frame 97 to frame 124, the double-bottom space from frames 97 to 103, out to the first longitudinal, being arranged for reserve feed. A small galley coal bunker is worked in between frames 59 and 63, on the starboard side, extending from the main deck down to the top of the settling tank.

The following are the particulars of the hull:

Length between perpendiculars, feet and inches	289–00
Projection forward of F.P., feet and inches	2–00
aft of A.P., feet and inches	2–08½
Length over all, feet and inches	293–10½
on L.W.L., feet and inches	289–04⅜
Breadth, molded, at 8 feet water line, feet and inches	25–11
10 feet water line, feet and inches	26–05
extreme, feet and inches	26–11⅞
Depth, molded, at frame 83, feet and inches	16–04 3/16
Draught, designed, feet and inches	8–04
Displacement at designed draught, tons	737.2
per inch at 8 inches draught, tons	12.2
Ratio of length to beam on W.L.	11.13
Area L.W.L. plane, square feet	5,032.5
wetted surface, square feet	7,770.0
immersed M.S., square feet	143.5
C.G. of L.W.L. plane aft of M.S., feet	2.78
C.B. above bottom of keel, feet	5.32
C.B. aft of M.S., feet	0.255
Transverse metacenter above C.B., feet	7.78
Longitudinal metacenter above C.B., feet	795.0
Coefficient of fineness, block	0.412
M.S.	0.663
L.W.L.	0.6699
Number of frames, spaced 21 inches	164.
Heights above 8-inch W.L.:	
Top of foremast, feet and inches	67–10¼
mainmast, feet and inches	68–03
bridge at side, feet and inches	31–00 1/16
forecastle deck at stem, feet and inches	25–04½
main deck at stern, feet and inches	16–04 3/16
searchlight platform, feet and inches	38–08 5/16

COMPLEMENT.

Commanding officer	1
Wardroom officers	3
Petty officers and crew	85
Total	89

SMALL BOATS.

Two 20-foot whale boats.
One 20-foot power dory.
One 17-foot dory.

MAIN ENGINES.

The propelling machinery consists of Parsons turbines, a description of which, covering the general features, is given below:

The arrangement provides for five ahead and two astern turbines, with the astern turbines in the casing with the L.P. ahead turbine. Three shafts are employed for going ahead and two for going astern, the disposition of the turbines being as follows: The center shaft is driven by the main H.P. turbine, the starboard shaft by the I.P. cruising, and one L.P. ahead turbine and the port shaft by the H.P. cruising and one L.P. ahead turbine. The cruising turbines for both the wing shafts are forward of the low-pressure turbines. For full speed ahead steam is admitted to the main H.P. turbine, and after expanding into the two L.P. turbines is exhausted into the condensers, the H.P. and I.P. cruising turbines under this condition revolving in a vacuum, maintained through the turbine drains.

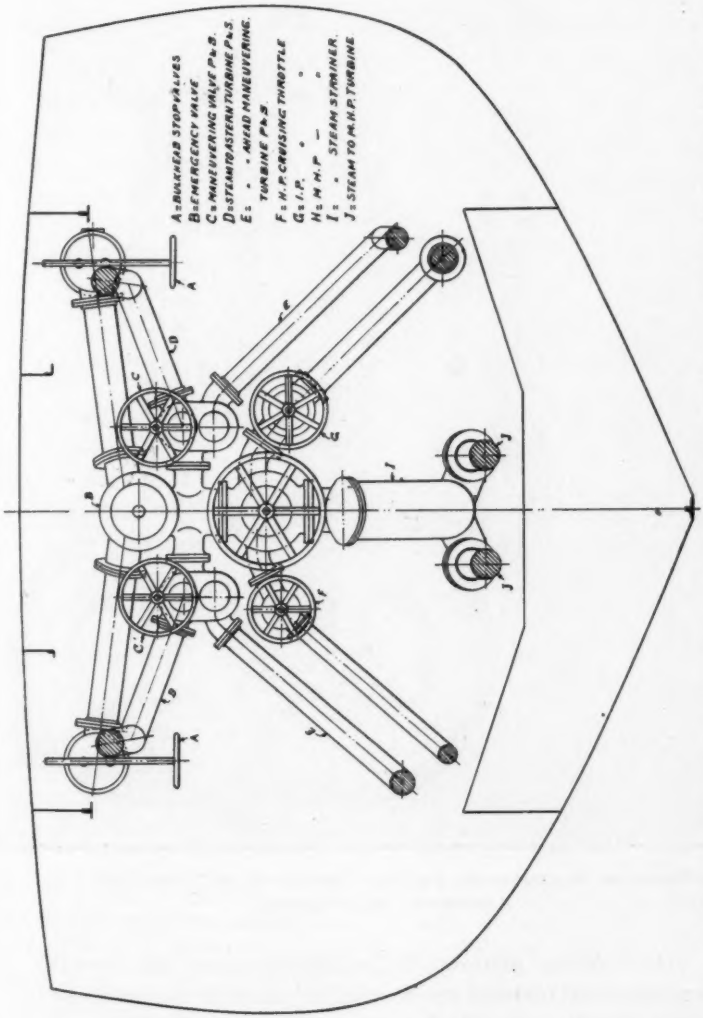
For low cruising speeds steam is admitted to the H.P. cruising turbine, expanded into the I.P. cruising turbine, and from this into the main H.P. turbine, thence through the L.P. turbines to the condensers.

For high cruising speed steam is admitted to the I.P. cruising turbines and expanded into the main H.P. turbine and thence

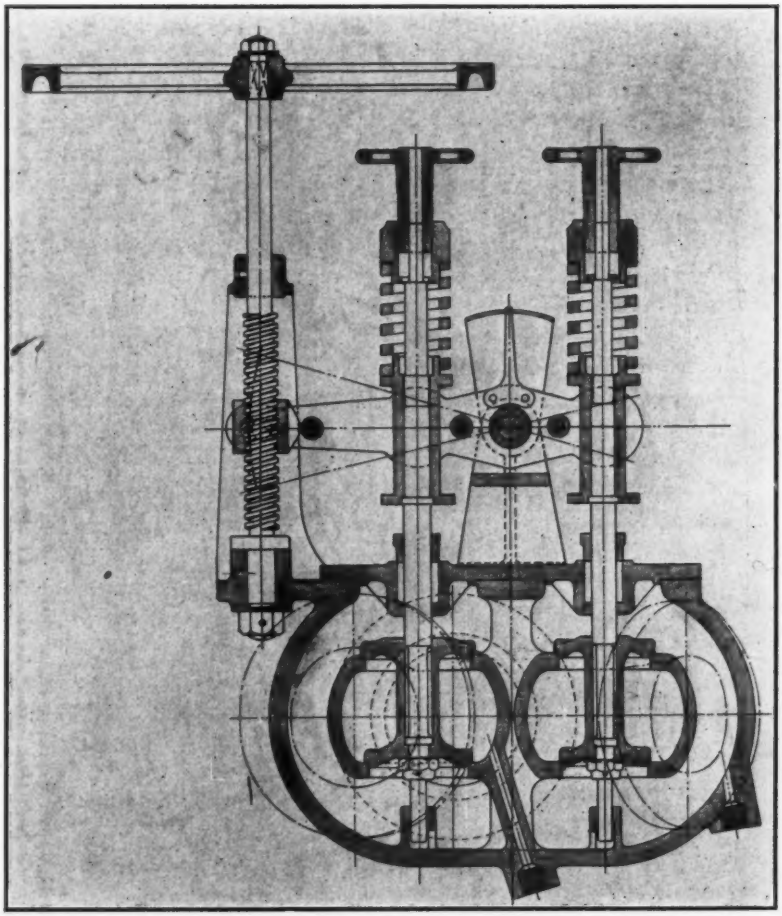
through the two L.P. turbines into the condensers. Steam is not admitted to the H.P. cruising turbine under this condition, the turbine revolving in a vacuum maintained through the drain, with the connection between the H.P. cruising and L.P. cruising turbines closed by a self-closing valve. Self-closing valves are also fitted on the exhaust pipe from the main H.P. to the L.P. turbines, so that the L.P. ahead and the astern turbines can be operated as desired, with the starboard and port engines entirely independent. Under these conditions the main H.P. turbine is not in use and is connected to the condenser by its drain. The exhaust pipes from the H.P. cruising to the I.P. cruising turbine and from the I.P. cruising to the main H.P. turbine are provided with screw-down self-closing valves.

The cylinders or casings are of hard close-grained cast iron, divided into two parts at the axis on a horizontal plane, the bearings at each end forming a part of the lower casting. This lower half is also provided with feet for bolting to the seatings. The method of machining the cylinders was to rough-bore and plane, after which the parts were bolted together and given the maximum hydrostatic test. Castings were then heated by steam at atmospheric pressure for 24 hours, to anneal same, and allowed to cool, after which the finishing work was done and the cylinders grooved for the blading.

The rotor drums are class A forged steel, and the rotor wheels, or spiders, are forged in one piece with the rotor shaft. The shafting is treated to give it the characteristics of class A forged steel, while the wheel portion is class B. These forgings were received at the works rough machined, with the spiders slotted out to their true proportions, thus giving a perfectly balanced rotor. This is a very expensive method, but undoubtedly gives most satisfactory results. The blading is on the Parsons standard system, the composition of the blades and binding strips 72 per cent. copper and 28 per cent. zinc; the calking strips 63½ per cent. copper and 36½ per cent. zinc, and the binding wire of copper.



ARRANGEMENT OF VALVE WORKING PLATFORM—TORPEDO-BOAT DESTROYERS "MCCALL," "AMMEN" AND "BURROWS."



SECTION THROUGH MANEUVERING VALVES—TORPEDO-BOAT DESTROYERS "MCCALL," "BURROWS" AND "AMMEN."

The working platform is immediately abaft the forward engine-room bulkhead, upon which is located all the valves and other operating gear, such as gages, counters, telegraphs, etc. Composing the group of turbine-operating valves are the 9-inch

angle, main steam stop valves, one on each side; the starboard valve admitting steam from the two after boilers, and the port valve from the two forward boilers. Pipes from these valves lead to a distributing casting located amidships, at the bottom of which is a 12-inch balanced throttle valve. At the bottom of the throttle-valve casting is a steam strainer with two 9-inch branches for steam to the main H.P. turbine, with a 7½-inch globe stop valve for steam to the I.P. cruising turbine on the starboard side, and a 5-inch globe stop valve for steam to the H.P. cruising turbine on the port side.

On the horizontal arms of the distributing casting are two combination double-beat valves for steam direct to the L.P. ahead and astern turbines, these valves being used for maneuvering; the valve stems being actuated by a double-ended lever controlled by springs, and so arranged that when one valve is opened the other is closed. The upper valve of the combination (see sketches) is 7-inch for steam to the L.P. astern turbines and the bottom valve is 6-inch for steam to L.P. ahead turbines. By turning the maneuvering-valve handwheel to the right the lower valve is closed and the upper one is opened, thereby admitting steam to the L.P. astern turbine, the operation, of course, being vice versa for going ahead.

All valves and castings are composition and all piping is of seamless-drawn steel, with rolled-steel flanges.

MAIN TURBINE DATA.

Diameter of rotor drums:	
Main H.P., inches	29½
H.P. cruising, inches	30
I.P. cruising, inches	29
L.P. ahead, inches	43½
L.P. astern, inches	29¾
Diameter of turbine casings:	
Main H.P., inches	32¾, 34, 35½, 38.
H.P. cruising, inches	30¾, 31, 31⅜.
I.P. cruising, inches	30¾, 31¼, 32.
L.P. ahead, inches	46¾, 48, 49½, 52, 54, 59.
L.P. astern, inches	31¼, 32½, 35.

Number of expansions:

Main H.P.	4
H.P. cruising	3
I.P. cruising	3
L.P. ahead	8
L.P. astern	4

Length of casing for blading, for each diameter and expansion noted above:

Main H.P., inches	13¼, 14⅞, 17, 18¼.
H.P. cruising, inches	16½, 17½, 17⅜.
I.P. cruising, inches	16¾, 21¾, 20½.
L.P. ahead, inches	7, 7⅛, 3 of 8, 8¼, 8⅜, 8½.
L.P. astern, inches	9¾, 11½, 14½, 15.

Rows of blading for each expansion:

Main H.P.	10
H.P. cruising	18
I.P. cruising	16
L. P. ahead	4 of 5, 4 of 4
L.P. astern	8

Length of blades for each expansion:

Main H.P., inches	1⅝, 2¼, 3, 4¾.
H.P. cruising, inches	0⅜, 0½, 0$\frac{11}{16}$.
I.P. cruising, inches	0⅞, 1⅛, 1½.
L.P. ahead, inches	1⅝, 2¼, 3, 4¾, 5¼, 3 of 7¾.
L.P. astern, inches	0¾, 1⅜, 2⅝, 2⅝.

Rotor shaft and bearings:	Length. *inches.*	Diameter. *inches.*	Diameter axial hole. *inches.*	Length over all of rotor drum and shaft. *feet.*	*inches.*
Main H.P.	7	7	3½	11	11$\frac{5}{16}$.
H.P. cruising	7	7	3½	10	02$\frac{3}{16}$.
I.P. cruising	7	7	3½	10	10⅝
L.P. ahead	12	7	3½	17	08$\frac{1}{16}$.
L.P. astern	12	7	3½	.....	

Thrust bearings:

Collars on shaft:	M.H.P.	H.P.C.	I.P.C.	L.P. (one).
Number	16	8	8	16
Thickness, inch	0½	0½	0½	0½
Space between, inch	0⅝	0⅝	0⅝	0⅝
Outside diameter, inches	8¾	8¼	8¼	8¾
Inside diameter, inches	5¼	5¼	5¼	5¼
Bearing surface, top, square inches	265.2	101.12	101.12	265.2
bottom, square in.	262.88	115.56	115.56	282.88
Number of shoes, top	15	7	7	15
bottom	16	8	8	16

Effective thrust surface (oil grooves out):

Full speed ahead, square inches	736.5
Astern, square inches	480.0
Cruising speed, square inches	736.5
Steam thrust, square inches	201.87+736.5=total, 938.37

Number and diameter of steam and exhaust connections:

	M.H.P.	H.P.C.	I.P.C.	L.P. ahead.	L.P. astern.
Steam, inches....	2 of 9	1 of 3	1 of 7½	1 of 6	1 of 7
Exhaust, inches..	2 of 16	1 of 9	1 of 13	1 of 39½×46⅝	

SHAFTING.

There are three lines of shafting, all of class A forged steel, arranged as follows: The starboard and port shaft abaft the turbines each consists of four sections, two of which are line, one stern-tube and one propeller. The center shaft consists of five sections, four of which are line and one stern-tube, the latter in this instance also being the propeller shaft. The shaft couplings are of the Lovekin type described many times in this JOURNAL. The stern-tube shafting is the only portion that is covered, the casing at the bearings being of composition H, and that between the bearings being seamless-drawn brass tubing ⅛-inch thick.

Data for Shafting.

Outboard shafting:	
Line shafting, forward section, length, feet and inches	12–01 7/16
after section, length, feet and inches	12–03
Stern-tube shafting, length, feet and inches	22–02½
Propeller shafting, length, feet and inches	17–02 11/16
Total length, feet and inches	63–09⅝
Center shafting:	
Line shafting, forward section, length, feet and inches	21–01 3/16
center, length, feet and inches	18–00
after center, length, feet and inches	16–06
section, length, feet and inches	14–00
Stern-tube, length, feet and inches	19–00 7/16
Total length, feet and inches	88–07⅝
Diameter of shafting, inches	06¾
axial hole, inches	03¾
coupling flanges, inches	11⅜
Thickness of coupling flanges, inches	01⅜
Diameter of shafting at steady bearings, inches	07⅛
Length of steady bearing, inches	09¼
Number of steady bearings:	
Outboard shafting, each	2
Center shafting	5
Length of inboard stern-tube bearings, feet and inches	1–09
outboard stern-tube bearings, feet and inches	2–09
strut bearings, feet and inches	2–09

PROPELLERS.

The propellers are of solid manganese-bronze, true screws, with the blades cast solid on the hub, which is bored taper to fit the propeller shaft, and held in place by two longitudinal keys and a composition nut on the end of the shaft. The nut is locked in place and covered by a composition cap bolted watertight to the hub. The driving faces of the blades were machined to a true surface, the backing faces being ground to a good surface, after which the entire surface was polished. Each propeller was accurately balanced. The starboard and center propeller are right-handed and the port left-handed.

Data for One Propeller.

Number of blades	3
Diameter of propeller, feet and inches	5–03¼
Pitch of propeller, feet and inches	4–10
Ratio of diameter to pitch	0.917
Area, projected, square feet	13.627
helicoidal, square feet	15.26
disc, square feet	21.82
Ratio of projected area to disc area	6.28
Area of immersed midship section, square feet	143.5
Ratio disc to immersed midship-section area	.151
projected disc to immersed midship-section area	.0949
Height of lower tip of blade above keel, wing, inches	01 9/16
below keel, center, feet and inches	1–04⅛
Immersion of upper tip of blade, wing, feet and inches	2–11¼
center, feet and inches	4–04⅞
Length of longitudinal keys, inches	13½
Thickness of longitudinal keys, inch	00¾
Width of longitudinal keys, inches	01½

LUBRICATION SYSTEM.

The main turbine and line-shaft bearings are provided with a closed system of lubrication, the oil being forced through by two pumps provided especially for this service. These pumps draw from a tank located amidships at the lowest available place in the engine room, and force the oil through one or both, if required, of the coolers, from whence it passes to the bearings at a pressure of about ten pounds per square

inch. It is thus collected by a series of pipes, uniting at a sight manifold, where a thermometer is provided for each bearing, so that the observer is able to note the temperature of the oil passing from each bearing and the fact that there is sufficient quantity as well. From this manifold the oil passes to the drain tank noted above, and the operation is thus made continuous. Reserve tanks are located under the main deck in the engine room, from which a supply of oil may be obtained to replenish that in the drain tank, should this be necessary on account of any leakage in the system. All piping is of copper or brass with composition fittings. For pumps and connections, see table of pumps.

The oil cooler used is the Schutte & Koerting corrugated-tube film oil cooler, consisting of nests of concentric spirally-corrugated copper tubes, forming by the differences in their diameters, "film spaces" through which the oil passes at a high velocity; the cooling water passing through the inside of the inner tubes and out through the space around the nest of tubes and the cooler shell. The tubes nest in "headers," that at the top being bolted to the shell flange and the bottom one free within the shell to permit of expansion. The cooling water and oil inlet and outlet are both at one end, this arrangement for the oil being accomplished by division walls in the headers to direct the flow. The shells are of copper, with composition flanges, heads, headers and tube nuts. The system will be recognized as that described in the article on "Heat Transmission and Transmitter," by Captain A. B. Willits, U. S. Navy, in the February, 1910, number of this JOURNAL.

The advantage of this type of cooler, over the plain-tube type, are increased cooling surface per axial foot, self-cleaning on account of the constant agitation due to the tube corrugation, facility in replacing a tube or in removing same for cleaning.

Constant agitation of the fluid to be cooled has the effect of bringing all of the particles in contact with the cooling surfaces, thus reducing to a minimum the size of the apparatus and the quantity of cooling agent used.

Data for One Cooler.	*McCall.*	*Burrows and Ammen.*
Number	2	1
Capacity per hour, gallons	2,000	2,960
From 150 degrees to 85 degrees, with water at 60 degrees.		
Length over all, feet and inches	4–04½	8–09½
of shell, feet and inches	3–03	7–07½
Diameter of shell, feet and inches	1–02	1–03
Number of tubes (nests)	6	3
Diameter of outer tube, I.D., inches	03	05 7/32
inner tube, O.D., inches	02¾	05 1/32
Gage of tubes, Stubbs, No	15	13
Length of outer tube, feet and inches	2–11	6–09⅜
inner tube, feet and inches	3–06	7–06⅜
Area of cooling surface, square feet	36	53.3
Water spaces tested to, pounds per square inch	50	
Oil spaces tested to, pounds per square inch	100	

TORSION METERS.

The torsion meters are of the Fottinger type, one for each shaft. Calibration of the shafts before installation gave the following data:

	McCall.			*Burrows.*			*Ammen.*		
	S	C	P	S	C	P	S	C	P
Length of shaft calibrated, feet	11	37–09	23–1 7/8	11	17	11	11	17	11
Foot-pounds for one-inch torque at 120-inch radius	11,428.55	3,373.34	5,458.08	11,666.66	7,466.66	11,428.55	11,570.247	7,427.055	11,546.39

The length of shaft which torsion meter records is for all three vessels, 4 feet 1.213 inches.

MAIN CONDENSERS.

There are two condensers, located one on each side of the engine room. They are cylindrical, with shells of wrought steel, water chests of copper, tube sheets of naval brass and tubes of composition 70 per cent. copper, 1 per cent. tin and 29 per cent. zinc.

The tubes are straight, fixed at one end by expanding into tube sheets and at the other end are provided with screw glands and packing.

The circulating water is furnished by independent circulating pumps discharging directly into the short water chest or nozzle at forward end of condenser, thence passing through the tubes and overboard through a similar nozzle.

Data for One Condenser.

Length of shell over tube sheets, feet and inches	12–02⅛
Inside diameter of shell, feet and inches	4–06½
Thickness of shell, inch	00¼
tube sheets, inch	01
Number of tubes	3,447
Outside diameter of tubes, inch	00⅝
Thickness of tubes, B.W.G. No	18
Length of tubes, feet and inches	12–02 1/16
Number of stay tubes	7
Pitch of stay tubes, inches	12¾
Size of stay tubes, steel pipe, inches, O.D	01⅜
gage, B.W.G	8
Cooling surface, measured on outside of tubes, square feet	4,804
Ratio of cooling surface to total heating surface	.5
Area through tubes, square inches	752
circulating water inlet and outlet, square inches	254.47
Ratio tube area to circulating inlet and outlet	2.9

The following connections are on each condenser shell:

One main turbine exhaust, rectangular, square feet	13.00
Auxiliary exhaust, inches	06
Bleeder, inches	02½
Feed-heater drain, inches	01¼
Air-pump suction, inches	06½
Safety valve, inches	02
Boiling-out connection, inch	00¾

The following connections are on each water chest:

One main injection on forward chest, inches	18
Outboard delivery on after chest, inches	18
Manhole, inches	12
Drain, inch	00¼
Air cock, inch	01
Thermometer, inch	01

MAIN AIR PUMPS.

There are two main air pumps of the Blake twin, vertical-beam type, having two 12-inch steam cylinders and two 26-inch water cylinders and a common stroke of 18 inches. Each is connected to its condenser by a 10-inch suction line and discharges to the main feed tank through a 9-inch pipe.

MAIN CIRCULATING PUMPS.

There are two main circulating pumps of centrifugal type, the 31-inch impeller being driven by a vertical simplex engine designed by the New York Shipbuilding Company, having enclosed crank case and forced lubrication.

Data for One Circulating Pump.

Capacity of pump, gallons per minute	10,000
Diameter of suction nozzle, inches	18
discharge nozzle, inches	18
impeller, inches	31
Width of tip of blades, inches	03½
Diameter of engine cylinder, inches	08
Stroke of engine cylinder, inches	07
Diameter of piston rod, inches	01¾

FEED AND FILTER TANK.

There is one feed and filter tank located on the center line of the ship at the after engine room and as high as possible to permit of entrance to the filter chamber. Total capacity is 650 gallons, of which 235 is for the filter chambers and 415 for the feed tank proper. The tank is of steel, galvanized. There are the following connections to the first filter chamber:

Two main air-pump discharge, inches	09
Filling pipe, inches	02
Distiller fresh water, inch	00¾
Vapor pipe, inches	01½
Drain pipe, inch	00¾

There are the following connections to the feed tank:

Main feed suction, inches	06
Overflow, inches	03
Vapor, inches	02
Drain, inches	01½

Water gauge and thermometer.

WATER-MEASURING TANKS—TRIAL.

A double tank was placed on deck over the main feed tank, fitted with gage glass and boards marked for each 10 gallons up to 730 gallons, the fair capacity of each tank. A temporary discharge from main air pump, with a swinging nozzle, was fitted up so that the air pumps could be caused to discharge to either of the contiguous measuring tanks at will. A large quick-opening valve in the bottom of the measuring tanks allowed them to be dumped into the main feed tank.

MAIN AND AUXILIARY FEED PUMPS.

There are two main feed pumps of the Blake vertical simplex piston type, located near the center line in the after part of the engine room under the feed and filter tank. They have 13-inch steam and 9-inch water cylinders, with a common stroke of 16 inches, a 6-inch suction to main feed and reserve tanks and a 4½-inch suction to the main air-pump channelways. They discharge to the feed line through a 5-inch line.

An auxiliary feed pump of same type and size is located in each boiler compartment, having a 4½-inch suction to main feed and reserve feed tanks, and discharge through a 4-inch line to boilers. These pumps are also provided with 1½-inch hose connections on both suction and discharge chambers.

FEED-WATER HEATER.

The feed-water heater is of the Schutte & Koerting type, similar to the oil cooler.

Data for Feed Heater.

Number	1
Capacity of water per hour (temperature range about 110 degrees F.), pounds	205,000
Length over all, feet and inches	8–09½
of shell, feet and inches	7–00½
Diameter of shell, feet and inches	2–03⅛
Number of tubes	8
Diameter of outer tube, I.D., inches	04½
inner tube, O.D., inches	04
Thickness, Stubbs, inches	10
Length of outer tube, feet and inches	7–03⅞
inner tube, feet and inches	6–01½
Area of heating surface (film), square feet	118
Total heating surface, square feet	126

BOILERS.

There are four boilers of the Thornycroft express type, oil-burning, two in each boiler compartment, each boiler having a separate smoke pipe. Each boiler has a single furnace.

Particulars and Dimensions of Boilers (Each).

Number of boilers (two in each boiler compartment)	4
Designed working pressure, pounds	250
Hydrostatic test applied, pounds	375
Steam pressure test applied, pounds	300
Combustion chamber, length, feet	8.7083
width, feet	11.3437
cubical contents, cubic feet	437
Number of oil burners, each boiler	11
Area through registers, square feet	10.56
casing air doors, square feet	14.9
Ratio	1.41
Height of pyrometers above floor, feet and inches	5–4
Number of furnaces	1
Heating surface (one boiler), area in square feet	4,800
Total heating surface, all boilers, in square feet	19,200
External height of boilers, feet and inches	10–03½
width of boilers, feet and inches	14–07⅞
length of boilers, feet and inches	12–07½
Tubes, outside diameter, inches	1⅛ and 1⅜
thickness, B.W.G., No.	10 and 11
kind	Cold-drawn seamless steel
number to each boiler (152=1⅜, 1905=1⅛)	2,057
longest, feet and inches	9–04¼
shortest, feet and inches	7–11⅞

Smoke pipes, height above grates, feet and inches	33–03
true ellipse, feet and inches	3–03×4–04
area of section, each, square feet	11
Drum, upper, diameter, inside, inches	42
thickness, top sheet, inch	$00\frac{9}{16}$
lower sheet, inches	$01\frac{3}{16}$
Upper drum heads are bumped to a radius of, inches	42
thickness, inch	$00\frac{3}{4}$
Lower drums, diameter, inside, inches	19
thickness, inch	$00\frac{9}{16}$

The oil-firing arrangements are the Schutte-Koerting patent system, an excellent description of which, with elevation and section drawings, will be found in "Notes on the Burning of Liquid Fuel," by Mr. Luther D. Lovekin, Chief Engineer of the New York Shipbuilding Company, in Volume XXIII, No. 1, 1911, "JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS."

OIL-MEASURING TANKS—TRIAL—"M'CALL" AND "BURROWS."

The settling tanks were fitted with gage glasses and boards marked to indicate the amount of oil in each tank. On the trials oil was used from opposite sides in the two firerooms to avoid listing the ship. When using oil from one side the tank on the opposite side was filled.

TRIAL "AMMEN."

Two steel tanks were located on the main deck between No. 2 and No. 3 smoke stacks. These tanks were fitted with a measuring tank of 100 gallons capacity in the top and a reservoir in the lower part, connected with a 4-inch quick-opening valve. A suction line led from the port tank to the fuel-oil service pumps in the forward boiler compartment, while the after pumps were supplied by the starboard tank. Counters were fitted to tally the number of times each measuring tank was dumped into its reservoir tank.

FORCED DRAFT.

The forced draft is supplied by two Terry turbines in each fireroom, direct connected to Sturtevant fans, 36-inch diameter for the *McCall* and *Burrows* and 39-inch diameter for the *Ammen,* designed to run at 1,500 r.p.m. Each blower is provided with a self-contained forced-lubrication system to each of its bearings. The blower shaft is vertical and its weight is taken by thrust ball bearings below the lower bearing, below which is a governor arranged to trip an emergency valve and prevent the turbine running away. This emergency valve can also be tripped by hand. Draft gages are fitted in each boiler compartment, to indicate the air pressure in inches of water.

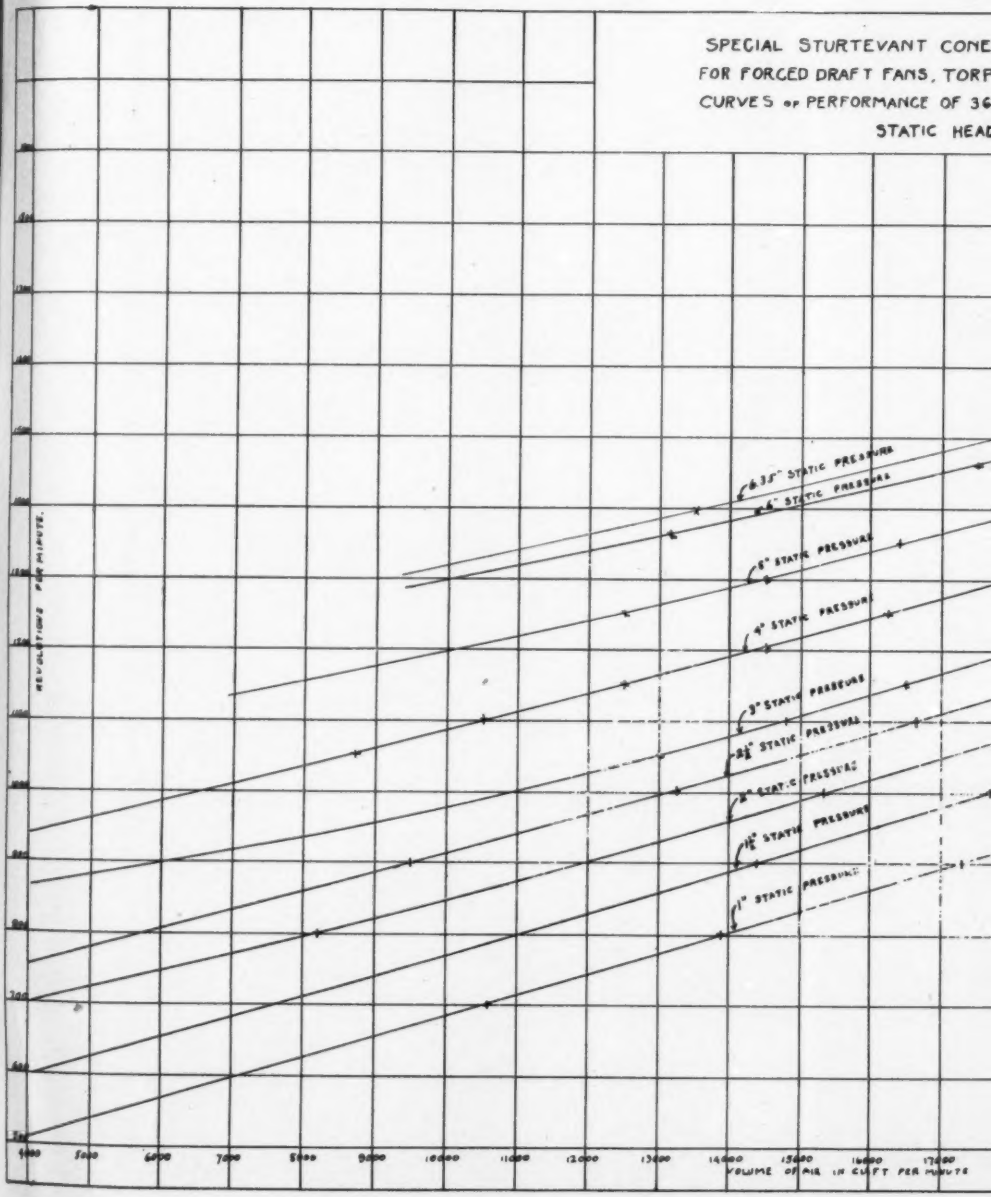
Tests of a fan unit representing those used on the *McCall* and *Burrows* were made and the results plotted on the curves shown on the accompanying chart.

EVAPORATING AND DISTILLING PLANT.

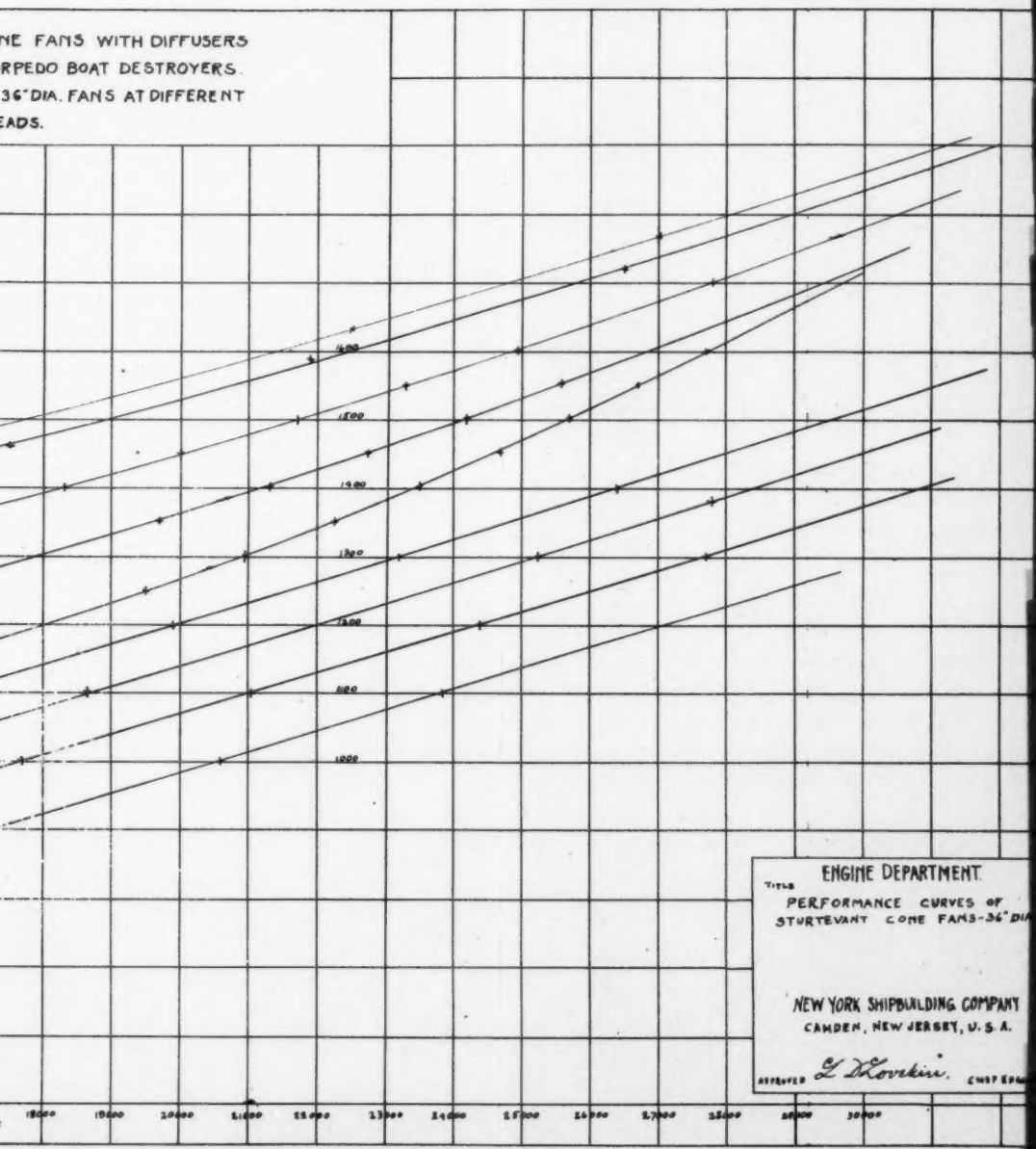
There are two Williamson Brothers evaporators, with a total heating surface of 150 square feet, located in the port after end of the engine room, and two Schutte & Koerting spirally-corrugated film-type distillers, located on the after engine-room bulkhead near the evaporators.

Data for One Evaporator.

Shell, inside diameter, feet and inches	2–04
Height, feet and inches	6–04
Number of tubes, U-shaped	40
Diameter, O.D., inches	01½
Thickness, Stubbs	13
Heating surface, square feet	75
Shell, working pressure, pounds per square inch	30
Coil, working pressure, pounds	250
Capacity, gallons in 24 hours	1,000


SPECIAL STURTEVANT CONE
FOR FORCED DRAFT FANS, TORF
CURVES OF PERFORMANCE OF 36
STATIC HEAD
REVOLUTIONS PER MINUTE.
6.35" STATIC PRESSURE
6" STATIC PRESSURE
5" STATIC PRESSURE
4" STATIC PRESSURE
3" STATIC PRESSURE
2½" STATIC PRESSURE
2" STATIC PRESSURE
1½" STATIC PRESSURE
1" STATIC PRESSURE
5000
5500
6000
7000
8000
9000
10000
11000
12000
13000
14000
15000
16000
17000
VOLUME OF AIR IN CU.FT PER MINUTE

NE FANS WITH DIFFUSERS
RPEDO BOAT DESTROYERS.
36" DIA. FANS AT DIFFERENT
EADS.



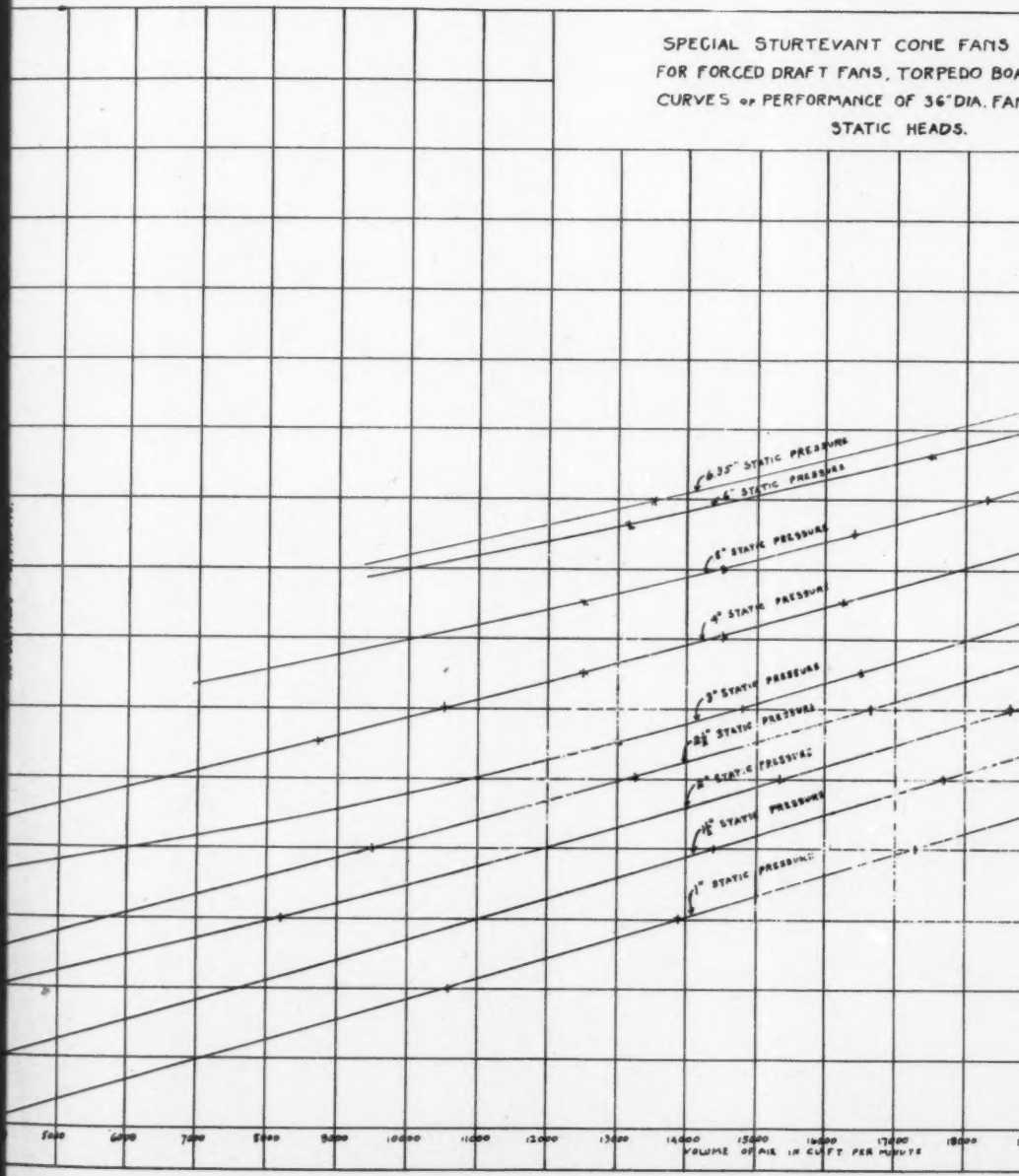
ENGINE DEPARTMENT

TITLE
PERFORMANCE CURVES OF
STURTEVANT CONE FANS-36" DIA

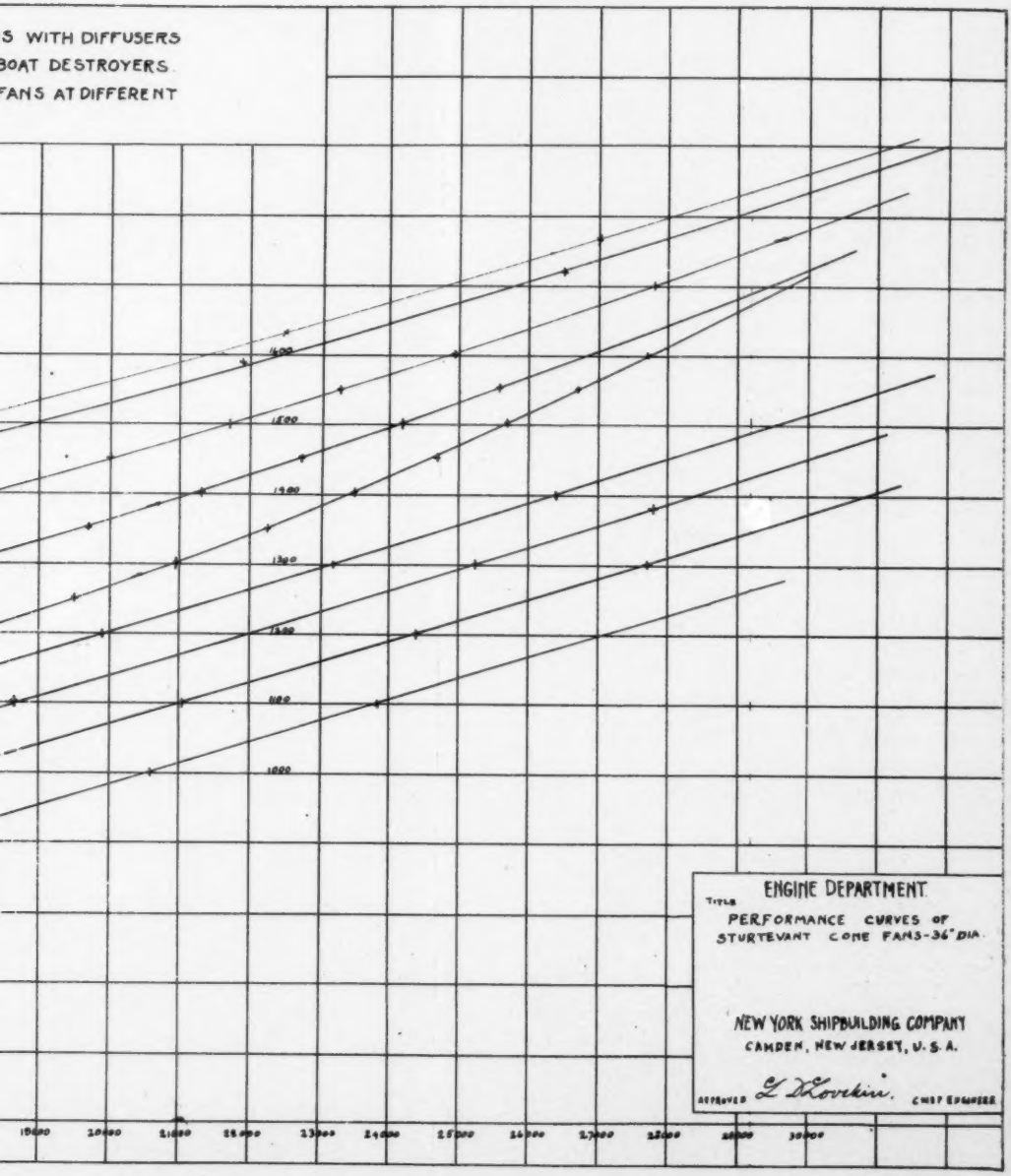
NEW YORK SHIPBUILDING COMPANY
CAMDEN, NEW JERSEY, U.S.A.

APPROVED L. L. Lovekin CHIEF ENG

SPECIAL STURTEVANT CONE FANS
FOR FORCED DRAFT FANS, TORPEDO BOA
CURVES OF PERFORMANCE OF 36" DIA. FA
STATIC HEADS.



S WITH DIFFUSERS
BOAT DESTROYERS
FANS AT DIFFERENT



ENGINE DEPARTMENT

TITLE
PERFORMANCE CURVES OF
STURTEVANT CONE FANS-36" DIA.

NEW YORK SHIPBUILDING COMPANY
CAMDEN, NEW JERSEY, U.S.A.

APPROVED L. Lovekin. CHIEF ENGINEER

Data for One Distiller.

Length over all, feet and inches	4–00
of shell, feet and inches	2–11¾
Diameter of shell, inches	07
Length of inner tube, feet and inches	3–02⅞
outer tube, feet and inches	2–11⅝
Thickness, Stubbs	15
Number of tubes, nests	1
Diameter outer tube, I.D., inches	03¾
inner tube, O.D., inches	03$\frac{23}{64}$
Cooling surface, film, square feet	6.1

All surfaces in contact with salt or fresh water are tinned. A four-hour test of the plant under service conditions gave the following average data:

	McCall.	*Burrows.*	*Ammen.*
Date of test	Dec. 7, 1910.	Jan. 6, 1911.	April 5, 1911.
Steam pressure in boilers	228	224	240.9
coils	191.5	155.4	138.7
shells	25.3	25.1	21.3
Average saturation	3/32	$\frac{2.26}{32}$	$\frac{3.2}{32}$
Temperature, external air	41.7	37	57
engine room	62.6	71.2	85.5
sea water	37.9	36	45
circulating-water discharge	38	51.1	51
distilled water from distillers	66.5	70.9	92.1
Gallons water distilled, 4 hours	360	365	401
Equivalent, 24 hours	2160	2190	2406

Quality water, *McCall*.....Fresh to 5 to 10 grains salt per gallon.
Burrows.....Fresh to trace.
Ammen.....Fresh—5 grains salt per gallon.

WINDLASS AND CAPSTAN.

The capstan is located on the forecastle with its engine secured immediately underneath on the watertight bulkhead at frame 12. It is a small vertical, double cylinder, steam engine, having cylinder of 4-inch diameter × 4-inch stroke, built by the Hyde Windlass Company, Bath, Maine, and is controlled by a lever on deck. On trial the following data was observed:

	McCall.	*Burrows.*
Number fathoms chain hove in	90	90
Time, minutes and seconds	10:40	11:31
Rate of heaving, fathoms per minute	8.25	7.8

STEERING ENGINE.

The steam-steering engine is located in the pilot house and is of the combined hand and steam type furnished by the Hyde Windlass Company, Bath, Maine. The steam cylinders are 5½ inches diameter by 5½ inches stroke.

There are three steering stations. A combined hand and steam gear in the pilot house, a steam gear on the bridge, and a hand gear aft on the main deck. All the steering gear is above the main deck. Transmission from the engine to the quadrant is by a ⅞-inch plough steel-wire rope, with ⅞-inch chains at sheaves. The rope is led along the main-deck gunwale through lignum-vitae guides spaced about ten feet apart. The rope ends aft in a sliding sheave through which the 1¼-inch chain from the quadrant leads, a single purchase being secured. The handwheel aft is connected to the leads by means of a turnbuckle fastened to a link of the chain.

Direct hand steering, by emergency gear, consists of a relieving tackle connected to the quadrant.

During the trials the gear was tested and results, as follows, obtained:

	McCall.	*Burrows*	*Ammen.*
Time to shift wheel on bridge, midship to hard-a-port, seconds	8.2	7.0	7.0
Time to swing 360 degrees change of course, minutes and seconds	3:08	2:53	2:57
Estimated diameter turning circle, yards	525	....	500
Initial angle of heel, degrees	3	....	....
Final angle of heel, degrees	1¾	....	....
Time to shift wheel on bridge hard-a-port to hard-a-starboard, seconds	15.02	15.	13.2
Time to swing 360 degrees change of course, minutes and seconds	3:14	2:45	3:04
Estimated diameter turning circle	500	....	500
Initial angle of heel, degrees	2½	....	....
Final angle of heel, degrees	1½	....	....
Time to shift wheel on bridge hard-a-starboard to midship, seconds	7	6	6.4

AUXILIARY CONDENSER.

There is an auxiliary condenser of the usual type, located in the starboard forward end of the engine room, worked by a Blake combined air and circulating pump. The condenser has 262 tubes, $\frac{5}{8}$-inch O.D., 5 feet $11\frac{3}{4}$ inches long, giving a cooling surface of 250 square feet.

PUMPS AND AUXILIARY ENGINES.

Auxiliary.	Number.	Type and make.	Steam cylinders.				Water cylinders.			
			Number.	Diameter.	Diam. piston rod.	Stroke.	Number.	Diameter.	Diam. piston rod.	Stroke.
				ins.	*ins.*	*ins.*		*ins.*	*ins.*	*ins*
Main air pumps	2	Blake, vertical, simplex, beam.	2	12	$1\frac{1}{2}$	18	2	26	2	18
Main circulating pumps	2	N. Y. S. Co., vertical, single-engine, centrifugal pump.	1	8	$1\frac{1}{2}$	7	1	31*		
Main feed pumps	2	Blake, vert., simplex, piston.	1	13	$2\frac{1}{2}$	16	1	9	2	16
Auxiliary feed pumps	2	Blake, vert., simplex, piston.	1	13	$2\frac{1}{2}$	16	1	9	$2\frac{1}{2}$	16
Engine-room fire and bilge pumps.	1	Blake, vert., simplex, piston.	1	10	$1\frac{7}{16}$	8	1	7	$1\frac{7}{16}$	8
Fireroom fire and bilge pumps.	2	Blake, vert., simplex, piston.	1	8	$1\frac{1}{2}$	12	1	5	$1\frac{1}{2}$	12
Auxiliary condenser pumps.	1	Blake, horizontal, simplex, combined air and circulating	1	6	$1\frac{1}{2}$	7	1 a 1 w	8 8	$1\frac{1}{2}$	7
Distiller fresh-water pumps	1	Blake, vert., simplex, piston	1	$3\frac{1}{2}$	$0\frac{3}{4}$	4	1	4	$0\frac{3}{4}$	4
Evaporator feed pumps	1	Blake, vert., simplex, piston.	1	$3\frac{1}{2}$	$0\frac{3}{4}$	4	1	4	$0\frac{3}{4}$	4
Lubricating-oil pumps	2	Blake, vert., simplex, piston.	1	6	$1\frac{7}{16}$	8	1	$5\frac{1}{4}$	$1\frac{7}{16}$	8
Oil-cooler circulating pumps.	1	Blake, vert., simplex, piston	1	6	$1\frac{7}{16}$	8	1	7	$1\frac{7}{16}$	8
Bilge pump	1	Blake, vert , simplex, piston.	1	$3\frac{1}{2}$	$0\frac{3}{4}$	4	1	4	$0\frac{3}{4}$	4
Fuel-oil pumps	4	Blake, vert., duplex, piston.	2	$4\frac{1}{2}$	$0\frac{13}{16}$	4	2	3	$0\frac{13}{16}$	4
Fuel-oil supply pumps	2	Blake, vert., simplex, piston.	1	6	$1\frac{7}{16}$	12	1	7	$1\frac{7}{16}$	12
Hand pump for fuel-oil system.	2	American Steam Pump Company, triplex.	...	...	...	...	3	$1\frac{1}{2}$	...	$3\frac{1}{2}$
Forced-draft blower engines	4	Terry Steam Turbine Company, single stage.								
Fans (*McCall* and *Burrows*).	...	36-inch Sturtevant, $29\frac{1}{2}$-inch diameter air inlet. Max. r p.m. 1,500.								
Fans (*Ammen*)	4	39-inch (as above).								
Air compressor, steam torpedo.	1	Capacity, 20 cubic feet at 2,500 pounds; revs., 375; H.P., 40; steam, 125; weight, 2,458 pounds.								
Dynamo turbines (*McCall* and *Burrows*).	2	Curtis, turbo-generator, single-stage, direct-connected, 5 kw., 5,000 r.p.m., 125 volts.								
Dynamo turbines (*Ammen*)	2	Terry turbine with Diehl generator, single-stage, direct-connected, 5 kw., 4,000 r.p.m., 125 volts.								

*Impeller: inlet, 18; outlet, 18.

PUMPS AND CONNECTIONS.

Pumps.	Ins.	Suction pipes from—	Ins.	Discharge pipes to—	Location.
Main air	10	Condenser	9	Filter tank	Engine room
Main circulating	18	Sea and bilge	18	Condenser	Engine room.
Main feed	4½	Feed tank and reserve feed tank.	4	Main feed	Engine room.
	4½	Air-pump channels			
Auxiliary feed	4½	Feed suction line and reserve feed tank.	4	Auxiliary feed line	One in each engine room.
	1½	Hose connections	1½	Hose connection	
Engine-room fire and bilge.	4	Sea	2½	Fire main	
	4	Drainage	3½	Overboard	
	1½	Hose connection	1½	Distiller flushing line	Engine room.
			1½	Hose connection	
Fireroom fire and bilge.	3½	Sea	2½	Fire main	One in each fireroom.
	3½	Drainage	3	Overboard	
	1½	Hose connection	1½	Hose connection	
Evaporator feed	1½	Sea	11	Evaporators	Engine room.
	1½	Distiller circulating discharge.			
Distiller fresh water.	1½	Distiller	1	Reserve tanks	Engine room.
			1	Ship's tanks	
Oil cooler, circulating.	3½	Sea	3	Oil cooler	Engine room.
Lubricating oil	3½	Lubricating-oil tank	3	Lubricating-oil supply line.	Engine room.
Fuel-oil service	2	Settling tanks	1½	Burners	Two in each fireroom.
Fuel-oil supply	4½	Storage tanks	4	Settling tanks	One in each fireroom.
Bilge	2	Drainage manifold	1½	Overboard	Under forecastle.
	1½	Hose connection			

ELECTRIC PLANT.

On the *McCall* and *Burrows* there are two Curtis turbo generators, single stage, direct connected, of 5-kw. capacity, at 125 volts, when making designed speed of 5,000 r.p.m., located forward on port side of engine room. The *Ammen* has Terry turbines—Diehl generators. A 1-kw. motor generator supplies current for the wireless-telegraph outfit. The switchboard is located near the generators, by which current is distributed to all parts of the vessel on the two-wire feeder system, and to an 18-inch searchlight located on a platform just above the bridge.

AIR-COMPRESSOR PLANT FOR TORPEDOES.

There is one torpedo air compressor, compressing in four stages, that will compress 20 cubic feet of free air to 2,500 pounds per square inch per minute, using steam at 125 pounds per square inch and making 375 r.p.m. There are ten navy-type accumulators with the necessary piping, gages, etc. An air line is taken into the firerooms and connections provided in engine room and firerooms with reducing valves for pneumatic tools and for cleaning boilers. The fuel-oil service pumps may also be operated by compressed air.

STEAM-HEATING SYSTEM.

The steam-heating system is of the usual navy type, with radiators of bent-brass pipe. Steam for heating forward compartments is taken off the auxiliary line in the forward boiler compartment, and that for the after heaters from the auxiliary line in after part of engine room.

DRAINAGE.

Bilge injection pipes and valves are fitted to both main circulating engines for removing large quantities of water from the engine compartment, while the ordinary leakage is removed by a line from the engine-room fire and bilge pump. Each of the fireroom fire and bilge pumps has a bilge suction for its own compartment only. There is no main or secondary drain as usually fitted.

Drainage of the forward compartments is accomplished by means of a small steam pump directly connected to two compartments and to others by means of a 1½-inch hose connection.

Drainage of the after compartments is entirely by means of hose connections to handy billy pump.

Drainage Scheme.

Valve.		Location.		Effect.
2-inch angle stop valve	A-2	Hold	C.L.	Sluice A-1
1½-inch globe stop valve, hose connection	A-15	Berthdeck	P.	Fill or drain A-2.
1½-inch deck-plate hose connection	A-16	Berthdeck	P.	Drain A-3.
2-inch manifold valve	A-16	Berthdeck	S.	Drain A-4.
2-inch manifold valve	A-16	Berthdeck	C.L.	Drain A-5.
1½-inch deck-plate hose connection	A-17	Berthdeck	C.L.	Drain A-7-M.
	A-17	Berthdeck	P.	Drain A-8-M.
	A-17	Berthdeck	P.	Drain A-9.
	D-11	Berthdeck	S.	Drain D-3.
	D-11	Berthdeck	S.	Drain D-4-M.
	D-11	Berthdeck	S.	Drain D-5-M.
	D-12	Berthdeck	S.	Drain D-6.
	D-12	Berthdeck	P.	Drain D-8.
1½-inch globe stop, hose connection	D-13	Berthdeck	S.	Fill or drain D-9.
2-inch angle sluice	D-9	Hold	S.	Sluice D-10.

VENTILATION.

Ventilation is by natural means in engine room and living space, 10-inch mushroom-type ventilators being provided for the living spaces, and a 40-inch cowl at the forward end of the engine-room hatch on the center line. The boiler spaces are sufficiently ventilated by the forced-draft blowers.

PIPING.

The main steam system comprises a 6½-inch lead from each boiler, the lines from the forward boilers joining and forming a 9-inch line running aft on the port side; the lines from the after boilers, joining and going aft on the starboard side, pass through 9-inch engine-room bulkhead stops and join at the nest of maneuvering valves at the working platform. Branches are taken off in each fireroom to supply steam to forward deck machinery, auxiliary feed pumps, fire and bilge pumps, forced-draft blowers, oil-supply and service pumps, etc.

In the engine room a 4-inch line is taken off to supply dynamo engines, circulating engines, main air and feed pump, air compressor, fire and bilge pump and evaporators. A 2½-inch bleeder line leads from the main steam line to both condensers. All steam pipes, 2 inches in diameter and above, are of seamless-drawn steel, with rolled-steel flanges; all below 2 inches are of copper with composition flanges.

The auxiliary exhaust line starts with a 2-inch pipe from the forward deck machinery and runs along the port side of the boiler compartments, picking up the exhaust from the various auxiliaries and increasing to 5 inches in the forward boiler compartment and 7 inches in the after boiler compartment, where a 6-inch line connects with the atmospheric exhaust. In the engine room the auxiliary exhaust forms a loop from port around to the starboard side, having a 6-inch connection to each main condenser, and a 3½-inch connection to the L.P. turbines, and ends at the auxiliary condenser on the starboard side, forward end of engine room.

WEIGHTS OF PROPELLING MACHINERY.

The weight of the propelling machinery is as follows:

	McCall.	*Burrows.*	*Ammen.*
	lbs.	lbs	lbs.
Main engines, casings, rotors, etc	94,415	94,703	96,275
Shafting (line and propeller), couplings	22,074	21,274	21,926
Line and propeller shaft bearings	3,684	4,291	3,639
Main condensers	34,827	35,749	34,311
Main air and circulating pumps	21,608	21,359	21,710
Propellers	4,181	4,171	4,328
Boilers	166,572	166,197	155,816
Boiler fittings	52,796	42,303	46,505
Smoke pipe, guys and uptakes	13,496	13,344	13,303
Steam and exhaust pipes and valves	40,040	39,671	41,084
Suction and discharge pipes and valves	21,889	21,867	24,878
Lagging and clothing	10,182	10,241	18,541
Flooring, gratings, etc	6,972	6,965	7,256
Auxiliaries	23,014	22,991	22,916
Fittings and gear	11,744	11,732	11,726

	McCall.	*Burrows.*	*Ammen.*
	lbs	lbs.	lbs.
Water	56,250	56,250	56,005
Stores, tools and spare parts	9,298	9,298	9,582
Miscellaneous machinery	46,674	46,628	50,772
Connections under steam engineering to other miscellaneous machinery	1,676	1,672	2,304
Total weight in pounds	641,392	630,706	642,877
Total weight in tons	286.335	281.565	286.998

REVERSING TESTS.

With the helm amidships and the vessel making approximately 29½ knots, the engine telegraphs were thrown "full speed astern," and the following data obtained:

	McCall.		*Burrows.*		*Ammen.*	
Time from signal to dead in water, minutes and seconds.	1:21		1:21		1:26	
Engines began turning over astern, seconds	S. 30.0	P. 34.5	S. 36.5	P. 43.6	S. 48.0	P. 47.0
Full speed astern, minutes and seconds	1:57	1:57	2:23	2:23	2:23	2:12

The indicators were then thrown full speed ahead.

Engines began turning over ahead, seconds	14.0	19.0	14.8	19.0	20.0	21.0
Full speed ahead, minutes and seconds	1:25	1:25	1:33	1:33	1:30	1:30

OFFICIAL TRIALS.

STANDARDIZATION TRIAL OF THE "M'CALL."

On September 6, 1910, standardization runs were made over the Delaware Breakwater measured course, near Lewes, Delaware. After several runs were made difficulty was had with the oil burners, and it was decided not to attempt the high-speed runs until some alterations were made. These were completed and a second set of runs made, beginning December 8, 1910. The weather was cool and pleasant with light haze on the horizon. Sea smooth and gentle breeze from W.S.W. The estimated displacement at the middle of the five high-speed runs was 721.7 tons.

U.S.S. McCALL
TRIAL Standardization.
COURSE Delaware Breakwater.
DATE December 8, 1912.
DATA
SCREW PROPELLERS
REVOLUTIONS PER MINUTE.
INDICATED, SHAFT AND EFFECTIVE HORSEPOWER.
SPEED IN KNOTS PER HOUR.
PROPULSIVE COEFFICIENT

U.S.S. BURROWS
TRIAL Standardization
COURSE Delaware Breakwater
DATE January 7, 1911
DATA
SCREW PROPELLERS
REVOLUTIONS PER MINUTE.
SPEED IN KNOTS PER HOUR.
INDICATED, SHAFT AND EFFECTIVE HORSEPOWER.
PROPULSIVE COEFFICIENT

U.S.S. AMMEN
TRIAL Standardization
COURSE Del. Breakwater
DATE April 6 1911
DATA.
SCREW PROPELLERS
REVOLUTIONS PER MINUTE.
SPEED IN KNOTS PER HOUR.
INDICATED, SHAFT AND EFFECTIVE HORSEPOWER.
PROPULSIVE COEFFICIENT

STANDARDIZATION TRIAL OF THE "BURROWS."

The *Burrows* was standardized over the Delaware Breakwater course on January 7, 1911. Weather cloudy but pleasant, with gentle W.N.W. wind, breeze changing to light westerly airs. Sea smooth, with short S.E. swells. Estimated displacement at the middle of the five high-speed runs was 727.5 tons.

On the standardization runs of both of these vessels the same turbine combinations were used for corresponding speeds as used on the four-hour full-speed trial, the ten-hour 25-knot trial and the ten-hour 16-knot trial. The speed and revolution curves were constructed from the data obtained, and from these curves the following means of the revolutions of the three shafts for various speeds obtained:

	McCall.	*Burrows.*	*Ammen.*
12 knots, r.p.m.	.....	.....	227.0
16 knots, r.p.m.	367.5	371.5	375.5
20 knots, r.p.m.	.....	.....	470.5
25 knots, r.p.m.	611.0	616.0	615.5
27½ knots, r.p.m.	.....	.....	694.0
29½ knots, r.p.m.	763.5	775.0	779.0

STANDARDIZATION TRIAL OF THE "AMMEN."

The propellers of the *Ammen* were standardized on the measured-mile course at the Delaware Breakwater, April 6, 1911. Twenty-six runs were made over the course. Five runs were made at the highest speed and three runs each at speeds of approximately 12, 16, 20, 22½, 25, 27½ and 29½ knots. During these runs the weather was pleasant, with light to moderate breezes. The mean displacement during the high-speed runs was 739.7 tons.

FUEL OIL.

The fuel oil used on these trials was a paraffine-base oil, which on analysis gave the following results:

	McCall.	*Burrows.*	*Ammen.*
Specific gravity at 45 degrees F.....	.....	.9002	.9005
50 degrees F.....	.8902	.....	.8965
55 degrees F.....	.8882	.8975	.8950
60 degrees F.....	.8865	.8958	.8935
65 degrees F.....	.....	.8935	.....
70 degrees F.....	.....	.....	.8900
72 degrees F.....	.8820	.....	.....
75 degrees F.....	.....	.8902	.....
85 degrees F.....	.....	.8860	.....

	McCall.	*Burrows.*	*Ammen.*
Flash, degrees F.....	300	273	288
Fire	320	...	...
Water and sediment, per cent.....under	1.	0.9	0.5
Reaction	Slightly acid.		Neutral.
B.T.U. per pound.....	19,618	19,598	19,503
Cold test, degrees F.....	5.	7.	0.0
Ultimate analysis, Sulphur, per cent.....	0.25	.34	.28
Hydrogen, per cent.....	12.96	12.52	12.84
Carbon, per cent.....	86.63	86.91	86.52

"M'CALL" STANDARDIZATION.

No. of Run.	St'b'd shaft. R.p.m.	S H.P.	Center shaft. R.p.m.	S.H.P.	Port shaft. R.p.m.	S.H.P.	Mean. R.p.m.	Total S.H.P.	Speed.
2	389.94	549	320.66	214	389.46	541	366.69	1,304	16.740
3	403.26	590	327.86	215	401.57	559	377.56	1,374	15.437
4	390.81	530	321.71	266	392.46	556	368.33	1,352	17.170
5	602.22	2,177	515.33	1,153	545.05	1,311	554.20	4,641	21.960
6	598.54	2,148	511.31	1,144	537.48	1,321	549.11	4,613	24.374
7	602.69	2,278	517.03	1,164	546.23	1,343	555.32	4,785	21.965
8	666.02	3,010	576.84	1,747	598.90	1,856	613.92	6,613	26.403
9	694.65	3,287	597.16	1,890	625.27	2,072	639.03	7,249	24.913
10	695.13	3,326	595.76	1,886	621.01	2,074	637.30	7,286	26.779
11	707.26	2,970	689.85	3,407	717.05	3,180	704.72	9,557	27.231
12	710.26	3,021	696.72	3,473	717.39	3,239	706.46	9,733	28.620
13	709.33	3,017	690.83	3,431	720.92	3,159	707.03	9,607	27.537
14	820.90	4,036	772.83	4,534	830.76	4,306	808.13	12,876	31.021
15	822.02	4,042	771.80	4,486	833.82	4,255	809.21	12,783	30.341
16	820.33	4,034	770.33	4,520	828.23	4,156	806.30	12,710	30.796
17	870.80	4,444	814.69	5,046	879.38	4,722	854.96	14,212	30.534
18	869.97	4,439	814.41	5,044	882.63	4,717	855.67	14,200	33.088
19	872.94	4,477	819.13	5,130	887.08	4,788	859.72	14,395	30.547
20	874.69	4,486	821.99	5,148	893.54	4,706	863.41	14,340	33.447
21	873.67	4,458	821.40	5,153	891.36	4,906	862.14	14,519	30.521

N.B.—Run No. 1 was thrown out on account of trouble with the chronograph.

"BURROWS" STANDARDIZATION.

No. of Run.	St'b'd shaft. R.p.m.	S.H.P.	Center shaft. R.p.m.	S.H.P.	Port shaft. R.p.m.	S.H.P.	Mean. R.p.m.	Total S.H.P.	Speed.
1	Chronograph stopped.								
2	294.23	231	231.50	63	291.00	226	272.24	520	10.980
3	294.88	231	232.23	57	291.62	235	272.91	523	13.096
4	292.45	222	231.72	63	289.56	225	271.24	510	10.692
5	393.63	543	321.03	193	390.50	524	368.39	1,260	17.405
6	390.55	539	321.31	193	386.63	519	366.16	1,251	14.252
7	386.50	512	318.05	191	381.31	501	361.95	1,204	17.044
8	491.12	1,148	418.56	594	441.46	652	451.05	2,394	17.939
9	500.41	1,205	419.76	574	452.23	692	457.47	2,471	20.674
10	498.17	1,187	420.24	586	450.55	689	456.32	2,462	18.432
11	593.16	2,071	501.65	1,040	533.34	1,145	542.72	4,256	23.524
12	589.67	2,043	499.68	1,036	528.57	1,135	539.31	4,214	21.710
13	587.61	2,019	498.07	1,033	527.76	1,119	537.81	4,171	23.236
14	667.33	3,116	588.32	1,863	621.58	2,018	629.08	6,997	24.645
15	680.49	3,113	594.38	1,914	626.50	2,034	633.79	7,061	26.233
16	679.30	3,125	592.67	1,893	624.60	2,045	632.19	7,063	24.846
17	702.53	3,004	674.20	3,165	712.43	3,097	696.39	9,266	28.151
18	697.70	2,964	670.37	3,147	710.16	3,106	692.74	9,217	26.659
19	704.32	3,012	674.55	3,167	713.65	3,141	697.51	9,320	27.926
20	Emergency valve closed partially and shut off steam.								
21	Observers could not see signals.								
22	810.84	4,060	759.89	4,418	825.87	4,211	798.87	12,689	29.185
23	806.22	4,037	757.45	4,404	824.97	4,206	796.21	12,647	31.034
24	812.48	4,090	761.19	4,425	829.35	4,206	801.01	12,721	28.987
25	875.01	4,736	815.37	5,163	891.25	4,831	860.54	14,730	32.794
26	881.56	4,772	820.25	5,149	894.88	4,731	865.56	14,652	30.346
27	873.14	4,750	813.72	5,108	889.06	4,772	858.64	14,630	32.744
28	881.43	4,795	820.50	5,173	896.03	4,809	865.99	14,777	30.270
29	879.44	4,808	819.56	5,190	895.04	4,804	864.68	14,802	33.084

"AMMEN" STANDARDIZATION.

No. of Run.	St'b'd shaft. R.p.m.	S.H.P.	Center shaft. R.p.m.	S.H.P.	Port shaft. R.p.m.	S.H.P.	Mean. R.p.m.	Total S.H.P.	Speed.
1	318.09	259	255.75	112	311.30	271	295.05	642	10.908
2	327.33	266	261.22	114	320.18	279	302.91	659	14.975
3	324.77	273	258.08	113	315.62	292	299.48	678	11.323
4	402.37	535	335.42	276	390.44	531	376.08	1,342	17.531
5	402.48	535	338.06	278	399.19	554	379.91	1,367	14.944
6	404.30	537	336.69	267	397.99	552	379.66	1,356	17.246
7	494.86	1,087	413.48	498	488.56	1,050	465.63	2,635	18.852
8	497.92	1,080	416.34	502	489.74	1,052	468.00	2,634	20,601
9	493.05	1,069	412.92	498	488.60	1,050	464.85	2,617	19.355
10	599.57	2,097	504.87	1,023	540.44	1,367	548.29	4,487	22.154

No. of Run.	St'b'd shaft. R.p.m.	S.H.P.	Center shaft. R.p.m.	S.H.P.	Port shaft. R.p.m.	S.H.P.	Mean R p.m.	Total S.H.P.	Speed.
11	600.95	2.167	504.85	1,023	539.36	1,364	548.39	4,554	23.543
12	605.22	2,182	507.78	1,057	543.82	1,361	552.27	4,600	22.336
13	679.60	3,114	583.92	1,696	609.74	2,023	624.42	6,831	26.019
14	683.16	3,223	585.04	1,795	613.62	2,052	627.27	7,070	24.407
15	680.25	3,190	584.60	1,762	611.39	2,028	625.41	6,980	26.451
16	720.54	3,203	689.61	3,287	728.47	3,447	712.87	9,937	26.744
17	723.51	3,197	690.06	3,252	728.66	3,428	713.91	9,877	29.377
18	715.63	3,162	686.64	3,198	724.23	3,427	708.83	9,787	26.461
19	801.70	3,934	746.68	4,050	804.81	4,268	784.40	12,252	30.981
20	797.85	3,915	745.72	4,045	808.54	4,288	784.04	12,248	28.206
21	793.99	3,856	745.73	4,045	806.27	4,300	781.83	12,201	30.946
22	890.47	4,804	828.18	4,991	889.19	5,175	869.28	14,970	30.117
23	887.36	4,835	825.21	5,019	885.69	5,154	866.09	15,008	32.871
24	886.66	4,928	824.52	5,014	886.17	5,157	865.78	15,099	29.915
25	899.25	4,998	834.71	5,031	883.13	5,198	875.70	15,227	33.349
26	899.74	5,098	838.16	5,143	899.84	5,335	872.25	15,575	30.293

FOUR-HOUR FULL-SPEED TRIALS.

"M'CALL."

The four-hour full-speed trial was begun at 9:10 A. M., December 9, 1910, the vessel then being about two miles E. by S. (mag.) from Five Fathoms Bank Lightship. The course out for two hours was E. by S. (mag.), weather fair, cool and pleasant, with moderate N.W. by N. breeze. Sea smooth with short chop. At 11:00 A. M. the ship was turned and headed back on course W.N.W. (mag.), but the breeze having freshened to fresh breeze with moderate, short, choppy sea, the high speed caused a heavy spray to be thrown completely over the deck with great force, making the recording of data very difficult. The ship was eased off until at 11:39 A. M. she was on course S.S.W. (mag.), and the trial was finished on that course at 1:10 P. M. with Cape Henlopen Light bearing about N.N.W. (mag.), distant about 98 miles. The mean displacement during this trial was 738 tons (estimated).

"BURROWS."

The four-hour full-speed trial was commenced at 9:55 A. M., January 8, 1911, Fenwick Island Lightship bearing N. by W. (mag.), distant about 7 miles. The course for the first two hours was E. by S. (mag.), and at 11:55 A. M. the ship was turned and the trial was completed on course N.W. by W. (mag.) at 1:55 P. M. Weather was fair, cool and pleasant, with gentle S.S.E. breeze, increasing to moderate to fresh breeze after the first hour. Sea was smooth with long, low swell from the Southeastward. Estimated displacement at the middle of the trial was 720.5 tons.

During the full-speed trials the following auxiliaries were in operation:

	McCall.	*Burrows.*
Main air pumps	2	2
Main circulating pumps	2	2
Main feed pumps	2	2
Fire and bilge pump (sanitary)	1	1
Fire and bilge pump (engine room)	1	1
	About half time.	
Forced-lubrication pumps	2	1
Oil-fuel pumps	2	2
Oil-tank pumps (occasionally)	2	1
Forced-draft blowers	4	4
Dynamo	1	1
Steering engine	1	1
Feed heater	1	1
Steam heating and galley	Yes.	Yes.
Boilers in use	4	4
Burners in use, A	8	10.4
B	9	10.9
C	11	11
D	11	11

PERFORMANCE FOUR-HOUR FULL-SPEED TRIALS.

Average steam pressures:	*McCall.*	*Burrows.*
At boiler gage, pounds	249.0	263.0
Engine room, main steam, gage, pounds	238.0	241.0
Auxiliary steam, gage, pounds	249.0	249.0
Auxiliary exhaust, gage, pounds	9.2	11.8
On glands, gage, pounds	1.5	1.03
Main H.P. turbine, absolute, pounds	192.0	195.0

	McCall.	*Burrows.*
H.P. cruising turbine, absolute, pounds	Idle.	Idle.
I.P. cruising turbine, absolute, pounds	Idle.	Idle.
L.P. turbine, starboard, absolute, pounds	44.3	43.9
port, absolute, pounds	44.3	41.6
Pressure in lubricating-oil system, pounds	7.85	9.38
Vacuum in condenser, starboard, inches of mercury	28.7	28.1
port, inches of mercury	28.2	28.0
Air pressure in fireroom, forward, inches of water	4.9	4.24
aft, inches of water	5.0	5.02
Pressure fuel oil, gage, pounds	148.2	127.5
Barometer	30.319	29.96
Revolutions, or double strokes, per minute:		
Shaft No. 1, starboard	824.0	839.302
2, center	772.0	774.227
3, port	833.0	856.522
Average	809.0	823.352
Pumps, main air, starboard	22.1	23.8
port	22.3	23.8
main circulating, starboard	274.7	304.4
port	274.4	309.4
main feed, starboard	26.0	23.9
port	26.3	23.9
forced lubrication	38.0	51.5
oil-cooler circulating	64.7	....
fire and bilge, engine room	24.0	33.8
fire room	44.7	64.0
fuel-oil service, starboard	39.3	48.8
port	49.8	50.1
Forced-draft blowers, average	1,391.0	1,441.0
Dynamos, ampères	30.1	31.2
Shaft horsepower:		
Shaft No. 1, starboard	4,024.0	4,334.0
2, center	4,477.0	4,487.0
3, port	4,571.0	4,433.0
Total	13,072.0	13,254.0
From standardization curve at revolution corresponding to trial speed	11,430.0	11,950.0
Indicated horsepower, auxiliaries	.....	420.
Collective horsepower, main engines and auxiliaries	.....	13,674.0
Speed of ship, knots	30.659	30.674
Slip of propeller, in per cent. of its own speed, based on mean pitch:		
Shaft No. 1, starboard	21.98	23.38
2, center	16.71	16.94
3, port	22.79	24.92
Mean	20.49	21.75

	McCall.	*Burrows.*
Temperatures, degrees Fahrenheit:		
Injection	50.9	46.5
Discharge, starboard	93.1	85.1
port	90.6	84.9
Air-pump discharge, starboard	....	85.0
port	....	84.9
Feed water, to heaters	87.2	84.9
from heaters	165.4	169.4
Outside air	....	47.9
Engine room working level	82.4	75.0
Fireroom working level, forward	66.7	89.2
aft	74.0	86.3
Main steam	401.0	400.0
Smoke-pipe gases	905.47	940.0
Lubricating oil from cooler	....	91.0
Fuel oil to burners	220.0	222.7
Consumption, water:		
Pounds per hour, all machinery	185,647.0	187,679.5
per S.H.P. main engines	14.202	14.1602
per H.P., all machinery	....	13.7252
Contract requirement	15.5	15.5
Actual consumption less than contract requirement	1.3	1.34
Fuel oil:		
Pounds per hour	15,080.9	14,507.4
Knots per ton (2240)	4.55	4.736

Deduced Data.

Square feet of heating surface per S.H.P.	1.469	1.449
Pounds fuel oil per S.H.P. per hour	1.154	1.095
H.P. per hour	....	1.061
Cooling surface (main condenser) square feet per S.H.P.	.735	.7249
Heating surface used during trial, square feet	19,200.	19,200.0
Pounds water evaporated per pound oil	12.31	12.937

TWELVE-HOUR TRIALS AT TWENTY-FIVE KNOTS.

"M'CALL."

The twelve-hour twenty-five-knot trial of the *McCall* was begun at 7:50 A. M., December 10, 1910, the vessel then being about 500 yards S.W. (mag.) from Five Fathom Bank Lightship. The course was E.N.E. (mag.); weather clear, cold and pleasant, wind increasing to fresh breeze from N. by

E. and short, moderate sea, sky becoming overcast and cloudy about 11:00 A. M., with light snow squall. Wind and sea moderated, and from 2:00 P. M. to end of trial there was smooth sea and calm to light N.W. airs. The trial ended at 7:50 P. M.

Draught at beginning of run, forward, feet and inches	8–5⅜
aft, feet and inches	8–7⅛
end of run, forward, feet and inches	7–6⅝
aft, feet and inches	8–7$\frac{13}{16}$
Estimated displacement at middle of 12-hour 25-knot trial	731 tons.

"BURROWS."

The Department having authorized the change from twelve hours to ten hours for the fuel-oil and water-consumption trial at twenty-five knots, the *Burrows* was run on that schedule and began her trial on January 10, 1911, at 8:30 A. M., the trial ending at 6:30 P. M. of the same day. The weather was clear, cool and pleasant, with fresh N.W. breeze and short, moderate sea from same direction, the wind moderating to calm to light westerly airs at end of the run, the sea going down with the wind.

Draught at beginning of run, forward, feet and inches	8–01½
aft, feet and inches	8–07⅜
end of run, forward, feet and inches	7–03$\frac{7}{16}$
aft, feet and inches	8–08$\frac{5}{16}$
Estimated displacement at middle of 10-hour 25-knot trial	720 tons.

PERFORMANCE, TWENTY-FIVE–KNOT TRIALS.

	McCall.	*Burrows.*
Duration of the trial, hours	12.0	10.0
Average steam pressures:		
At boiler, gage, pounds	252.0	260.0
Engine room, main steam, gage, pounds	248.0	248.2
Auxiliary steam, gage, pounds	252.0	249.0
exhaust, gage, pounds	9.25	8.3
On glands, gage, pounds	1.5	1.0
Main H.P. turbine, absolute, pounds	94.7	105.0
H.P. cruising turbine, absolute, pounds	Idle.	Idle.
I.P. cruising turbine, absolute, pounds	245.0	242.9
L.P. turbine, starboard, absolute, pounds	21.0	20.9
port, absolute, pounds	22.8	21.1

	McCall.	*Burrows.*
Pressure in lubricating-oil system, pounds	6.9	8.95
Vacuum in condenser, starboard, inches of mercury	29.7	29.2
port, inches of mercury	28.6	29.1
Air pressure in fireroom, forward, inches of water	3.55	4.06
aft, inches of water	3.55	4.25
Pressure fuel-oil gage, pounds	120.0	104.7
Barometer	30.34	30.284
Revolutions, or double strokes per minute:		
Shaft No. 1, starboard	672.0	670.5
No. 2, center	591.0	597.1
No. 3, port	606.0	625.9
Average	623.0	631.3
Pumps, main air, starboard	22.9	22.3
port	22.8	22.3
circulating, starboard	225.0	260.0
port	230.0	270.0
feed, starboard	31.0	16.4
port	....	18.5
forced-lubrication	32.7	51.0
oil-cooler circulating	58.2	....
Pumps, fire and bilge, engine room	27.9	31.4
fireroom	52	54
fuel-oil service, starboard	36.8	24
port	52	23.4
Forced-draft blowers, average	1,185	1,069
Dynamos, ampères	23	32.2
Shaft horsepower:		
Shaft No. 1, starboard	3,018	3,017
No. 2, center	1,912	2,066
No. 3, port	2,164	2,324
Total	7,094	7,407
From standardization curve at revolutions corresponding to trial speed	6,430	6,490
Indicated horsepower, estimated, auxiliaries	....	275
Collective horsepower, main engines and auxiliaries	....	7,682
Speed of ship, knots	25.369	25.459
Slip of propeller, in per cent. of its own speed, based on mean pitch:		
Shaft No. 1, starboard	20.82	20.4
No. 2, center	9.93	10.61
No. 3, port	12.30	14.72
Mean	14.35	15.24
Temperatures, degrees Fahrenheit:		
Injection	46.7	41.8
Discharge, starboard	82.6	76.8
port	83.6	73.5

	McCall.	*Burrows.*
Air-pump discharge, starboard	....	68.9
port	....	68.9
Feed water, to heaters	74.4	68.9
from heaters	184	184.3
Outside air	....	39
Engine room working level	76	82.4
Fireroom working level, forward	61.9	78
aft	70.23	80.4
Main steam	404.	....
Smoke-pipe gases	828.	685.27
Lubricating oil from cooler	72.	....
Fuel oil to burners	206	....
Consumption water:		
Pounds per hour, all machinery	104,316.6	108,954.8
per S.H.P. main engines	14.705	14.7097
H.P., all machinery	....	14.181
contract requirement	16.75	16.75
Actual consumption less than contract requirement:		
Fuel oil, pounds per hour	8,131.2	8,790.5
knots per ton (2,240)	6.99	6.487

Deduced Data.

	McCall.	*Burrows.*
Square feet of heating surface, per S.H.P.	2.031	2.592
Pounds fuel oil per S.H.P. per hour, all machinery	1.147	1.187
H.P. per hour	....	1.144
Cooling surface (main condenser), square feet per S.H.P.	1.354	1.297
Heating surface used during trial, square feet	14,400.	19,200.
Pounds water evaporated per pound oil, square feet.	12.829	12.395

Auxiliaries in Use.

	McCall.	*Burrows.*
Main air pumps	2	2
circulating pumps	2	2
feed pumps	2	2
Forced-lubrication pump	1	1
Oil-cooler circulating pump	1	...
Fire and bilge pump, flushing	1	1
engine room	1	1
Fuel-oil service pumps	2	2
supply pumps	2	1
Dynamo	1	1
Feed heater	1	1
Forced-draft blowers	3	4
Steering engine	1	1
Heating system	Yes.	Yes.
Boilers in use	3	4

Burners in use:	*McCall.*	*Burrows.*
Boiler "A"	9	6.5
"B"	9	7
"C"	5 to 8	6
"D"	....	6

TWELVE-HOUR TRIALS AT SIXTEEN KNOTS.

"M'CALL."

The twelve-hour sixteen-knot trial of the *McCall* was begun at 8:05 A. M., December 11, 1910, and ended at 8:05 P. M. of the same day. The course was between Overfalls Lightship to Five Fathoms Bank Lightship, and back and forth to Fourteen Foot Bank. The weather was overcast, cloudy, cold and misty, with frequent snow squalls, fresh N. to N.E. breezes, sea smooth with short chop.

Draught at beginning of run, forward, feet and inches	8–01
aft, feet and inches	8–08½
Draught at end of run, forward, feet and inches	7–09$\frac{15}{16}$
aft, feet and inches	8–09¼
Estimated displacement at middle of 12-hour 16-knot trial	738.61 tons.

"BURROWS."

The duration of the *Burrows* sixteen-knot trial having been changed from twelve to ten hours, it was begun at 8:35 A. M., January 9, 1911, and ended at 6:35 P. M. the same day. The course was back and forth between Overfalls Lightship and Fourteen Foot Bank Light. Weather was clear, cool and pleasant, fresh N.W. by N. breeze, which freshened to moderate gale at 11:30 A. M., with short, choppy sea.

Draught at beginning of run, forward, feet and inches	7–11½
aft, feet and inches	8–09
Draught at end of run, forward, feet and inches	7–10
aft, feet and inches	8–08⅜
Estimated displacement at middle of 10-hour 16-knot trial	735.2 tons.

PERFORMANCE, SIXTEEN-KNOT TRIAL.

	McCall.	*Burrows.*
Duration of trial, hours	12.0	10.0
Average steam pressure at boiler, gage, pounds	240.0	254.0
Engine room main steam gage, pounds	234.0	229.0
Auxiliary steam gage, pounds	237.0	234.0
exhaust, gage, pounds	7.2	7.3
On glands, gage, pounds	1.3	1.0
Main H.P. turbine, absolute, pounds	....	26.1
H.P. cruising turbine, absolute, pounds	184.0	196.5
I.P. cruising turbine, absolute, pounds	66.3	75.2
L.P. turbine, starboard, absolute, pounds	7.3	7.55
port, absolute, pounds	8.4	7.5
Pressure in lubricating-oil system, pounds	6.3	7.95
Vacuum in condenser, starboard, inches of mercury.	28.8	29.0
port, inches of mercury	28.6	28.52
Air pressure in fireroom, forward, inches of water	....	....
aft, inches of water	3.2	4.45
Pressure, fuel oil, gage, pounds	121.5	135.8
Barometer	30.17	29.93
Revolutions, or double strokes, per minute:		
Shaft No. 1, starboard	384.0	395.47
No. 2, center	333.0	330.93
No. 3, port	383.0	389.69
Average	367.0	372.03
Pumps, main air, starboard	24.0	25.0
port	24.0	....
circulating, starboard	190.6	240.0
port	184.5	255.0
feed, starboard	....	10.3
port	16.3	....
forced-lubrication	29.4	34.7
oil-cooler circulating	29.5	....
fire and bilge, engine room	....	....
fireroom	62.4	55.5
fuel-oil service, starboard	....	....
port	16.6	20.9
Forced-draft blowers, average	1,426.0	1,388.0
Dynamos, ampères	25.6	31.8
Shaft horsepower:		
Shaft No. 1, starboard	496	579
No. 2, center	266	295
No. 3, port	625	569
Total	1,387	1,443
From standardization curve at revolutions corresponding to trial speed	1,350	1,320
Indicated horsepower, estimated, auxiliaries	....	134

	McCall.	*Burrows.*
Collective horsepower, main engines and auxiliaries	....	1,577
Speed of ship, knots	15.97	16.031
Slip of propeller, in per cent. of its own speed, based on mean pitch:		
Shaft No. 1, starboard	12.79	15.02
No. 2, center	0.0	1.55
No. 3, port	12.66	13.75
Mean	8.48	9.07
Temperatures, degrees Fahrenheit:		
Injection	39.1	34.8
Discharge, starboard	60.3	48.8
port	60.4	48.4
Air-pump discharge, starboard	....	56.9
port	....	56.9
Feed water, to heaters	58.6	56.9
from heaters	216.3	215.9
Outside air	29	35.8
Engine room working level	81	71.7
Fireroom working level, forward	....	....
aft	69	78.5
Main steam	402.7	....
Smoke-pipe gases	732	702
Lubricating oil from cooler	65.1	....
Fuel oil to burners	217.8	217.1
Consumption, water:		
Pounds per hour, all machinery	32,887.8	33,132.4
per S.H.P. main engines	23.711	22.961
H.P., all machinery	....	21.01
Contract requirements	27	27
Actual consumption less than contract requirements	3.289	4.039
Fuel oil:		
Pounds per hour	2,614.5	2,575.3
Knots per ton (2,240)	13.68	13.944

Deduced Data.

Square feet of heating surface per S.H.P.	3.46	3.3264
Pounds fuel oil per S.H.P. per hour	1.885	1.785
H.P. per hour, all machinery	....	1.633
Cooling surface (main condenser), square feet, per S.H.P.	6.297	6.658
Heating surface used during trial, square feet	4,800	4,800
Pounds water evaporated per pound oil	12.579	12.865

Auxiliaries in Use.

Main air pumps	2	1
circulating pumps	2	2
feed pumps	1	1

	McCall.	*Burrows.*
Forced-lubrication pumps	1	1
Oil-cooling circulating pumps	1	1
Fire and bilge pumps, flushing	1	1
engine room	1	1
Fuel-oil service pumps	1	1
supply pumps	1	1
Dynamo	1	1
Feed heater	1	1
Forced-draft blower	1	1
Steering engine	As required.	
Heating system	Yes.	Yes.
Boilers in use	1	1
Burners, boiler "C"	7	..
"D"	..	6.6

TRIALS OF THE "AMMEN."

The trials of the *Ammen* consisted of the standardization trials above noted, and a series of 4-hour runs in the open sea at speeds of approximately 12, 16, 20, 25½, 27½ knots, and the highest speed attainable. During these runs the conditions were as nearly as practicable the same as on the standardization trials at corresponding speeds.

DATA FOR FOUR-HOUR RUNS.

Date and time of trial	Apr. 10, A. M.	Apr. 12, A. M.	Apr. 11, P. M.	Apr. 10, P. M.	Apr. 11, A. M.	Apr. 7.
Displacement, mean, on trial, tons	747.5	737.7	737.4	730.3	738	735.9
Speed of ship, in knots per hour	11.835	15.898	19.795	25.541	27.659	30.483
State of sea	long swell	short chop	short chop	long swell	long swell	long swell
Revolutions of main shafts:						
Starboard	295	395	494	694	708	842
Center	235	322	407	582	679	782
Port	289	396	495	621	712	844
Mean revolutions	273	371	465	632	700	823
Slip of propellers, in per cent. of their speed:						
Starboard	15.89	15.61	15.99	22.84	18.09	24.27
Center	(−) 5.59	(−) 3.52	(−) 1.97	7.99	14.59	18.27
Port	14.14	15.83	16.16	13.77	18.55	24.45
Mean slip	8.15	9.31	10.06	14.87	17.04	22.33
Steam pressures:						
Boilers, gage	241.7	240	260	252.1	250	250.1
Main steam pipe at throttle	240.4	239.3	256.3	252	249.8	246.7
Auxiliary steam pipe	240	237.6	251.8	252	249	248.7
exhaust	8.5	9.09	8.1	12.34	9.9	10
On glands	.3	.32	.14	.2	.24	.49
Main H.P. steam chest	15	20	35.3	85	148.6	205.9
H.P. cruising steam chest	66.1	155.4	249.5	Idle.	Idle.	Idle.
I.P. cruising steam chest	28.5	65.8	105	250	Idle.	Idle.
L.P. turbine chest, starboard	5.7	8	11.5	21.2	32.9	46.3
port	5	9	12.5	21.2	33.4	46
Pressure in lubrication system	5	6.6	6.3	6.5	7.25	8
Vacuum in condenser, starboard	29	29	29.7	29.1	28.8	27.3
port	29	29	29.1	29.1	28.6	27.9
Air press. in firerooms, ins. water	3.58	4.43	4.09	3.90	5	5.84
Barometer	30.26	30.33	30.29	30.24	30.31	29.76

DATA FOR FOUR-HOUR RUNS—Continued.

Auxiliaries:						
Revolutions or double strokes:						
Main air pumps, starboard	22	22	20	22	22	24
port	...	...	20	22	22	24
circ. pumps, starboard	247	273	325	260	275.6	325
port	247	275	286.3	266	274.1	325
feed pumps, forward	9.6	11.88	16.2	17.2	20.2	26.5
aft	...	...	...	12.3	20.2	26.5
Fuel-oil pumps, forward	...	...	...	35.3	35.9	58.1
aft	12.8	16.31	24.1	18	31.9	42.2
Forced-lubrication pumps	32.9	37.19	46.8	43.1	46.6	58
Fire and bilge pumps	...	...	...	25	30	34.5
Forced-draft blowers	1,035	1,244.4	1,024	1,098	1,133.7	1,293
Dynamo watts	3,876	3,552	3,540	3,864	3,605	3,546
Number of boilers in use	1	1	2	3	4	4
Average number of burners in use, each boiler	5	8.34	5.65	9.03	8.8	11
Pressure of oil to burners	124.6	136.8	117	104.55	115.95	128.4
Temperatures, Fahrenheit:						
Fuel oil to burners	206	218.9	215.3	220.85	219.45	221.75
Injection	41	42.5	47.1	43 4	41	42.7
Discharge	55.3	57.8	60.5	74.95	80.25	81.55
Air-pump discharge	70	...	...	62.4	...	85.75
Feed	70	72.3	85.3	62.4	79.6	162.7
Outside air	45.8	46.1	44.2	47 2	45.5	51.1
Engine-room working level	78	73.5	79.4	80	71.4	77.4
Fireroom working level	86.9	80.3	80	82 45	80.7	83.75
Horsepower, main shafting:						
Starboard	227	559	1,104	3,460	3,201	4,450
Center	45	194	368	1,702	3,108	4,649
Port	376	709	1,301	2,291	3,431	4,902
Total	648	1,462	2,773	7,453	9,740	14,001
Water:						
Pounds per hour, all machinery	19,438.15	32,870	47,791.74	102,510.24	145,932.6	195,224
Pounds per hour per H.P. of main shafts, all machinery	29.97	22.483	17.235	13.754	14.982	13.944
Fuel oil:						
Analysis and heating power	Ultimate analysis, C. 86.52; H. 12.84; S. 0 28; B.T.U.'s 19,503					
Pounds per hour, all purposes	1,844	3,013	4,366	9,104	12,888.5	16,264
knot run	155.8	189.5	220.6	356.4	466	534.2
Knots per ton	14.38	11.82	10.16	6.28	4.81	4.19
Deduced data:						
Sq. ft. of heat. surface per S.H.P.	7.4074	3.283	3.462	1.9321	1.971	1.371
Pounds fuel oil per S.H.P. pr. hr.	2.846	2.061	1.574	1.222	1.323	1.163
burner pr. hr.	369.	360.8	386.4	335 9	366.1	370.1
Pounds of water evaporated per pound of fuel oil	10.541	9.31	10.946	11.260	11.323	11.989
Cooling surface (main condenser) per S.H.P.	14 827	6.572	3.4648	1.2891	.986	.6862
Auxiliaries in operation:						
Main air pumps	1	1	2	2	2	2
circulating pumps	2	2	2	2	2	2
feed pumps	1	1	1	2	2	2
Forced-lubrication pumps	1	1	1	1	1	1
Fire and bilge pumps	2	2	2	2	2	2
Auxiliary-feed pumps	Not used.	Seldom.	1 occasionally.	1 occasionally.	1 occasionally.	Very little.
Oil-tank pumps	1 occasionally.	1 very little.	1	1	1	1
Oil-service pumps	1	1	1 occasionally.	2	2	2
Forced-draft blowers	1	1	2	3	4	4
Dynamos	1	1	1	1	1	1
Steering engine	As required.					
Feed heaters	Not used.	Not used.	Not used.	Not used.	Not used.	1
Steam heaters	Very little.	Moderately.	Moderately.	Very little	Moderately.	Very little
Steam to galley	As required.					

POST TRIAL EXAMINATION.

After the trials the following machinery parts were opened up, disassembled and examined:

	McCall.	*Burrows.*	*Ammen.*
Boilers	4	4	4
Turbine casings	All	All	All
Main air pump	1	1	1
feed pump	1	1	1
Fire and bilge pump	1	1	1
Auxiliary feed pump	1	1	1
Oil-service pump	1	1	1
Main circulating engine	1	1	1
Thrust bearings	All	All	All
Blower engine	1	1	1
Condensers	3	3	3
Feed heaters	..	..	1

On the *McCall* it was found that the boiler zincs required renewal. The tips of casing blades showed rubbing on rotor drums, but none of the blades were bent. Three blades were broken loose from the brazing to binding strip on one L.P. rotor. The balance of the machinery was found to be in excellent condition.

A few rotor blades were found not properly soldered to the binding strips on the *Burrows,* and there were slight leaks in the safety valves, the balance of the machinery being in excellent condition.

On the *Ammen* a few fire brick were found to be cracked, and a few of the steam-gland rings on the I.P. cruising-rotor spindle and cap were found damaged. Feed-water heater was found leaking, due to nuts holding corrugated tubes having worked loose. Balance of machinery was found in excellent condition.

Throughout the trials the fuel-oil-burning apparatus worked satisfactorily and the smoke was never excessive, varying in color from a light grey to moderate dark and seemed well under the control of the operators. The high-speed fan system for forced draft caused a rush-of-air noise that could be heard at a considerable distance, and under favorable wind and weather conditions might be audible for some miles.

U. S. TORPEDO-BOAT DESTROYER *TRIPPE*.

By Commander U. T. Holmes, U. S. Navy.

CONTRACT TRIALS.

The *John Trippe* is one of the five torpedo-boat destroyers authorized by Act of Congress approved March 3, 1909. The contract for her construction was entered into with the Bath Iron Works, Bath, Maine, on June 15, 1909. The keel was laid April 12, 1910, and the vessel was launched practically complete December 20, 1910.

The *Trippe* is in all essential particulars a replica of the *Paulding* and *Drayton*, which were fully described in the November, 1910, number of the Journal. A detail description will not be included with this account of the trials of the vessel.

SCHEDULE OF TRIALS.

The *Trippe* was subjected to the following trials:

(1) Standardization trials over the measured mile at speeds of about 12, 15, 18½, 22½, 25, 27½, 29½ and the highest speed attainable. Three runs were made at each speed except the highest, and five runs at the highest speed.
(2) A four-hour full-speed trial.
(3) A four-hour 27½-knot trial.
(4) A four-hour 25-knot trial.
(5) A four-hour 20-knot trial.
(6) A four-hour 16-knot trial.
(7) A four-hour 12-knot trial.

The consumption of fuel oil and the amount of water used was determined by actual measurement on each of the six four-hour trials. The arrangements for measuring oil and water were as follows:

For oil three tanks of about 100 gallons capacity each were placed on deck in wake of the boiler compartments. The tank-supply pumps were arranged to deliver to these tanks and the oil-pressure pumps were arranged to take their suction direct from them. Cocks were placed in the suction from and discharge to each tank, enabling a shift to be made instantly when a tank was filled or emptied. The temperature of the oil was taken at each filling of a tank and the final results were corrected for the difference in weight corresponding to the observed temperatures. The tanks were carefully calibrated before the trials.

For water three tanks of about 400 gallons capacity each were placed over the engine room. The air pump delivered to these tanks through quick-closing valves and there was provision for letting down the contents into the suction pipe to main feed pumps for feeding the boilers. When a tank filled, the air-pump discharge was instantly shifted to the empty tank, and when a tank was emptied, the feed-pump suction was shifted to a filled tank. By the use of three tanks there was never any confusion or difficulty in regulating either the air-pump discharge or feed-pump suction.

These tanks were carefully calibrated before the trial. The temperature was taken for each tank of water and the necessary corrections have been applied in the tabulated data.

The standardization trials were run over the measured-mile course off Rockland, Maine, during the forenoon of February 21, 1911. Twenty-nine runs were made, but runs Nos. 13, 14 and 15 were thrown out on account of failure to get the time correctly on run No. 15.

The same combinations of turbines were used on the standardization trials as were to be used on the

Four-hour full-speed trial,
Four-hour 27½-knot trial,
Four-hour 25-knot trial,
Four-hour 16-knot trial,
Four-hour 12-knot trial.

The twenty-knot speed on the standardization trial was run with the four-turbine combination, but the four-hour 20-knot trial was run with the five-turbine combination.

At the conclusion of the standardization trials the usual helm and other maneuvering trials and tests of the anchor gear were made with satisfactory results.

From the results of the standardization trials a curve of speeds and revolutions was constructed, from which it was deduced that the mean of the revolutions of the three shafts for the various speeds required on the four-hour trials was as follows:

At 29½ knots, 806.0 revolutions per minute.
At 27½ knots, 719.5 revolutions per minute.
At 25 knots, 634.0 revolutions per minute.
At 20 knots, 485.0 revolutions per minute.
At 16 knots, 380.5 revolutions per minute.
At 12 knots, 285.0 revolutions per minute.

STANDARDIZATION TRIALS.

February 21, 1911, a. m.

No. of run.	Mean r.p.m. three shafts.	Total S.H.P. three shafts.	Mean speed, knots.
1	286.05	581.	13.092
2	280.70	618.	10.824
3	281.56	594.	12.778
4	361.73	1246.	14.400
5	375.67	1353.	16.572
6	366.45	1344.	14.686
7	440.35	2108.	19.064
8	445.04	2400.	17.751
9	459.59	2530.	19.882
10	558.08	4737.	21.918
11	563.09	4817.	23.392
12	563.27	4878.	22.215
16	631.68	7007.	24.395
17	633.49	7003.	25.483

No. of run.	Mean r.p.m. three shafts.	Total S.H.P. three shafts.	Mean speed, knots.
18	632.37	7057.	24.434
19	713.84	9757.	27.821
20	710.16	9877.	26.826
21	716.29	9936.	27.832
22	802.43	12734.	28.946
23	809.95	12877.	30.000
24	805.71	12869.	29.114
25	877.56	15402.	31.405
26	874.57	15354.	30.822
27	884.19	15548.	31.565
28	880.81	15485.	30.087
29	884.58	15488.	31.537

The various four-hour trials were conducted on February 21st to 24th, with results as given in the appended table. The curves of speed and revolutions, speed and horsepower of main turbines and propulsive coefficients are also appended.

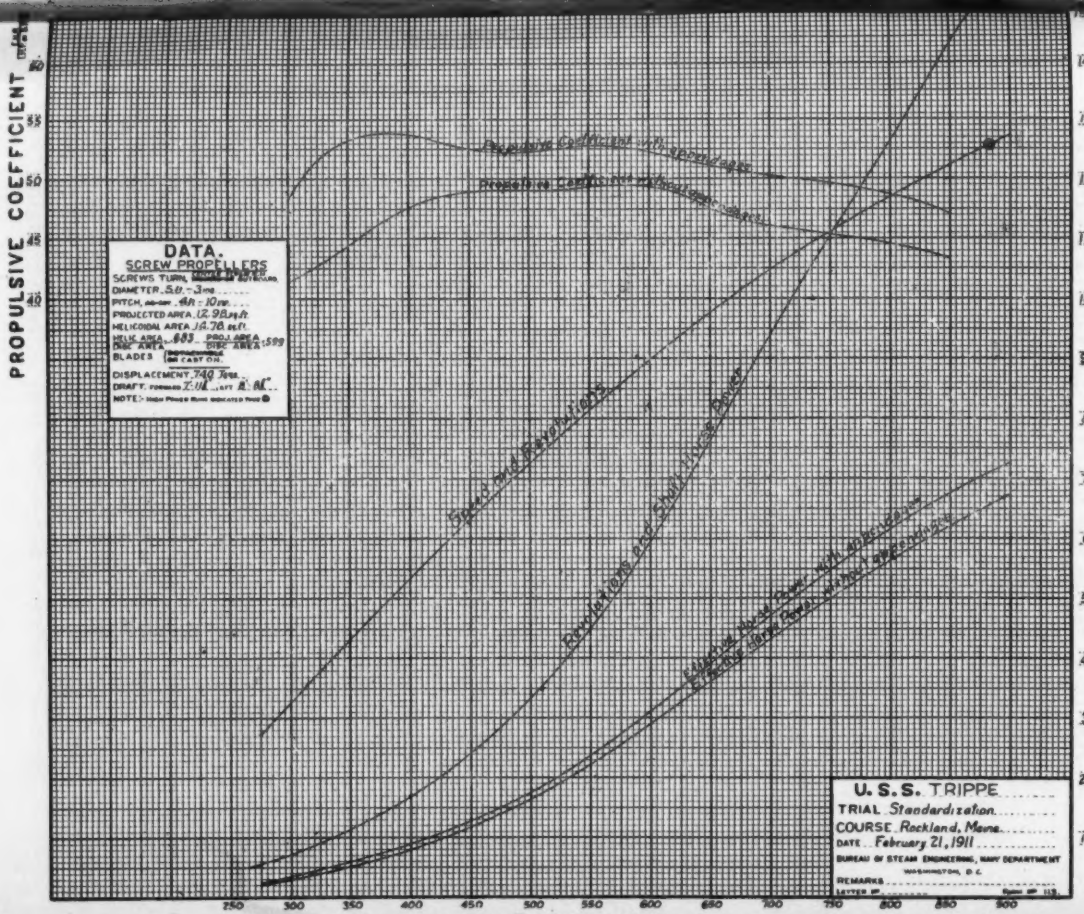
The trials of this vessel are noteworthy in that they are the first in our Navy in which reliable data was obtained concerning fuel and water consumption at various speeds between the lowest cruising speed and the maximum speed of the vessel.

DATA FOR FOUR-HOUR RUNS.

Date and time of trial	Feb. 22, A. M.	Feb. 24, A. M.	Feb. 23, P. M.	Feb. 23, A. M.	Feb. 22, P. M.	Feb. 21, P. M.
Mean draught on trial, ft. and ins	8-04	8-02½	8-02$\frac{1}{16}$	8-03$\frac{1}{2}$	8-02½	8-03½
Corresponding displacement, mean draught, tons	739	726	725	730	722	733
Trim by stern, inches	7¼	14	13$\frac{7}{8}$	10$\frac{1}{2}$	12$\frac{1}{16}$	11$\frac{7}{8}$
Speed of ship, in knots per hour	11.461	16.095	19.724	25.185	27.744	30 891
State of sea	Smooth and short chop.	Mod., confused.	Short chop to mod.	Short chop.	Short chop.	Short chop.
Revolutions of main shafts:						
Starboard	285.1	406.09	501.16	682.18	734 70	880.58
Center	249.2	339.61	433.43	607.78	705 41	834.97
Port	282 9	403.86	497.64	630.29	748.54	890.69
Mean revolutions	272.6	383.18	477.41	640.08	729.55	868.78
Slip of propellers, in per cent. of their speed:						
Starboard	15 72	16.90	17.48	22.60	20 83	26.46
Center	3.57	.64	4 59	13 12	17.54	22.43
Port	15.06	16 44	16.90	16.22	22.29	27.28
Mean slip	11.45	11.327	12.99	12.985	20 22	25.39
Steam pressures:						
Boilers, gage	177	226	248	259	226.3	250 5
Main steam pipe at throttle	176	207	248	259	222 0	249
Auxiliary steam pipe exhaust	170.5	221	238	254	218.3	241
	4	2.1	3.4	8	6.3	5
On glands	1	1	1	1	1	1
Main H.P. steam chest	15	30	50	121	185	250

DATA FOR FOUR-HOUR RUNS—Continued.

Steam pressures:						
H.P. cruising steam chest	68.9	165	240	Idle.	Idle.	Idle.
I.P. cruising steam chest	34.7	75	115	256	Idle.	Idle.
L.P. turbine chest, starboard	5	10	14.8	28	41 4	57
port	6	11	15.4	30	43.0	57
Pressure in lubrication system	3.34	3.4	4.5	5	4.4	8.5
Vacuum in condenser, starboard	28.97	28.5	28.5	28.5	29.0	28.5
port	28.97	28.5	28.5	29.5	28.0	27
Air press. in firerooms, ins. water	1.6	2.9	1.96	2.65	2.8	4.1
Barometer	29.54	29.32	29.52	29.67	29.46	29.68
Auxiliaries:						
Revolutions or double strokes:						
Main air pumps, starboard	26.7	33.4	...	37	36.4	40
port	...	...	45.3	45	35.6	38
circ. pumps, starboard	77 1	123.2	210	265	350	395
port	73	100.1	169	273	315	394
feed pumps, forward	14	...	21.1	16	24	26
aft	...	12.2	...	17	24	33
Fuel-oil pumps, forward	10.3	213	24.8	43	43.6	61
aft	...	...	...	18	38.8	71
Forced-lubrication pumps	15.3	21	22	18	44	53
Fire and bilge pumps	18.6	29.8	34.5	41	35 6	37
Dynamo engines	...	...	...	...	...	...
Forced-draft blowers	735	F. 1,083	F. 898 3	F. 1,448.4 A. 826.2	1,263	1,462
Number of boilers in use	1	1	2	3	4	4
Average number of burners in use, each boiler	3	7	4.5	7	7.35	10
Pressure of oil to burners	191.3	290.9	286.6	280	236.7	280
Temperatures, Fahrenheit:						
Fuel oil to burners	209.4	197.5	207.2	200	196 3	196
Injection	32.4	37	33.5	35.4	33 9	34
Discharge	53.2	{ 60 { 63	53.4 56 1	63.2 61.9	66.9 69.2	74 76
Air-pump discharge	64 9	67	{ 56.1 { 56.4	65.2 66.6	64.0 64.1	67 68
Feed	220.7	207	201	205 6	199.3	179
Outside air	...	29	24	16.6	24.5	19
Engine-room working level	71.3	78	68.5	71	70.5	65
Fireroom working level	48	52.1	57.9	54.3	61.7	57.5
Horsepower, main shafting:						
Starboard	170	584	1,110	3,129	3,368	4,766
Center	257	472	820	2,336	3,819	5,494
Port	130	475	954	1,943	3,206	4,718
Total	557	1,531	2,884	7,408	10,393	14,978
Water:						
Pounds per hour, all machinery	18,617.5	35,833.4	51,939.6	110,348	150,681.75	206,006.75
Pounds per hour per H.P. of main shafts, all machinery	33.425	23.405	18.01	14.896	14.498	13.754
Fuel oil:						
Analysis and heating power	Ultimate analysis: C, 86.64; H, 12.97; S, 0.39. B.T.U.'s by Mahler's calorimeter, 19,607.					
Pounds per hour, all purposes	1,456	2,826	3,941	8,881	11,641	17,304
knot run	125.4	175.6	199.8	352.6	419.6	560.2
Knots per ton	17.632	12.758	11.721	6.352	5.339	3.999
Deduced data:						
Sq. ft. of heat. surface per S.H.P.	8.672	3.155	3.350	1.956	1.859	1.290
Pounds fuel oil per S.H.P. pr. hr.	2.614	1.846	1.367	1.199	1.120	1.155
burner pr. hr	485.3	403.7	437.9	422.9	396.0	432.6
Pounds of water evaporated per pound of fuel oil	12.787	12.680	13.179	12.425	12.944	11.905
Cooling surface (main condenser per S.H.P.	17.239	6.272	3.330	1.296	0.924	0.641
Auxiliaries in operation:						
Main air pumps	1	1	1	2	2	2
circulating pumps	2	2	2	2	2	2
feed pumps	1	1	1	2	2	2
Forced-lubrication pumps	1	1	1	1	1	1
Fire and bilge pumps	1	1	1	1	1	1
Auxiliary-feed pumps	Occasionally as required.					
Oil-tank pumps	1	1	1	1	1	1
Oil-service pumps	1	1	1	2	2	2
Forced-draft blowers	1	1	1	2	2	2
Dynamos	1	1	1	1	1	1
Steering engine	Occasionally as required.					
Feed heaters	1	1	1	...	1	1
Steam heaters	Yes.	Yes.	Yes.	Yes.	Yes.	No
Steam to galley	Yes.	Yes.	Yes.	Yes.	Yes.	Yes.


PROPULSIVE COEFFICIENT
INDICATED, SHAFT AND EFFECTIVE HORSEPOWER.
SPEED IN KNOTS PER HOUR.
REVOLUTIONS PER MINUTE.
Propulsive Coefficient
Speed and Revolutions
Indicated Horsepower
Effective Horsepower
DATA.
SCREW PROPELLERS
DIAMETER 5.0'-3 ins
PITCH, mean 4ft-10 ins
PROJECTED AREA 12.98 sq ft
HELICOIDAL AREA 14.78 sq ft
HULL AREA .883 PROJ AREA .589
DISC AREA .589
DISPLACEMENT 740 Tons
U. S. S. TRIPPE
TRIAL Standardization
COURSE Rockland, Maine
DATE February 21, 1911
BUREAU OF STEAM ENGINEERING, NAVY DEPARTMENT
WASHINGTON, D. C.
REMARKS

THE U. S. TORPEDO-BOAT DESTROYER *WALKE*.

By U. T. Holmes, Commander, U. S. N.

CONTRACT TRIALS.

The *Henry Walke* is one of the five torpedo-boat destroyers authorized by Act of Congress approved March 3, 1909. The contract for her construction was entered into with the Fore River Shipbuilding Company, Quincy, Massachusetts, June 15, 1909. The keel was laid March 5, 1910, and the vessel launched November 3, 1910.

The *Walke* is in all essential particulars like the *Perkins* and *Sterett*, which were described in the February, 1911, number of the Journal. The trial data for the *Ammen* and *Trippe*, two other vessels of the same date, appear elsewhere in this number.

SCHEDULE OF TRIALS.

The *Walke* was subjected to the following trials:

(1) Standardization trials over a measured mile, at speeds of about 12, 14, 16, 20, 22½, 24½, 25½, 27, 29½ knots and the highest speed attainable. Three runs were made at each speed, except the highest, and five runs at the highest speed.

(2) A four-hour full-speed trial.

(3) A four-hour 27½-knot trial.

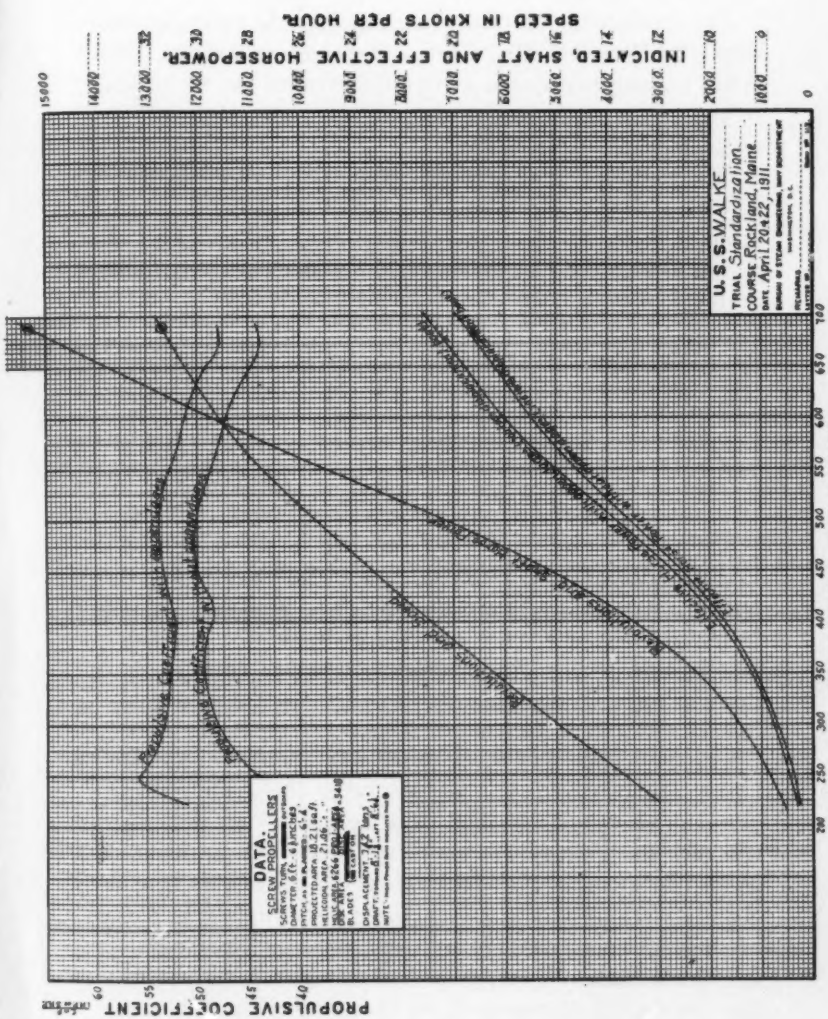
(4) A four-hour 25-knot trial.

(5) A four-hour 20-knot trial.

(6) A four-hour 16-knot trial.

(7) A four-hour 12-knot trial.

The consumption of fuel oil and the amount of water used were determined by actual measurement on each of the six four-hour trials. This was done by employing temporary tanks placed on deck for the purpose. These tanks were carefully calibrated before the trial. The temperature was


U. S. S. WALKE
TRIAL Standardization.
COURSE Rockland, Maine.
DATE April 20 & 22, 1911.
DATA.
REVOLUTIONS PER MINUTE.
INDICATED, SHAFT AND EFFECTIVE HORSEPOWER.
SPEED IN KNOTS PER HOUR.
PROPULSIVE COEFFICIENT

taken for each tank of water or oil and the necessary corrections have been applied in the developed data.

The standardization trials were run over the measured-mile course off Rockland, Me. They were begun on April 20, 1911. After making twenty-one runs, thick weather set in, making it impossible to see the ranges. These trials were resumed and completed on April 22d.

The results of the standardization trials are given in the following table. A curve of speeds and revolutions was constructed from which it was deduced that the mean of the revolutions of the two shafts for the various speeds required on the four-hour trials was as follows:

29½ knots	611.2
27½	549.2
25	494.
20	377.6
16	297.7
12	223.3

The various four-hour trials were conducted on April 20th to 24th, 1910. The sixteen-knot trial, which was first run on April 21st, was at the request of the contractors, repeated on April 22d, on account of irregularity of the revolutions on the first trial. This permitted a better showing to be made in water consumption per hour per shaft horsepower of the main propelling turbines.

The four-hour full-power trial was conducted on April 24th. The results of this and the other four-hour trials are given in the following table. The figures for the full-power trial are the average for the four hours, but it should be noted that at the end of the first two hours of this trial the engines were slowed to give a speed very slightly in excess of the 29½ knots required by the contract. This was done in order to make the best possible showing in water consumption per shaft horsepower of the main propelling turbines.

The curves of speed and revolutions and horsepower of main turbines and propulsive coefficients are appended hereto.

STANDARDIZATION DATA.

No. of runs.	Shafts.				Average R.p.m.	Speed.	Total S.H.P.
	Starboard.		Port.				
	R.p.m.	S.H.P.	R.p.m.	S.H.P.			
1	264.862	495	258.042	470	261.452	14.691	965
2	253.138	446	253 775	448	253.457	13.027	894
3	252.410	426	247.772	410	250.091	14.090	836
4	229.700	341	230.823	344	230.262	11.717	685
5	235.790	336	233.791	330	234.791	13.255	666
6	215.765	282	218.294	289	217.030	11.020	571
7	296 182	679	294.296	670	295.239	16.393	1,349
8	296.010	695	297.332	701	296.671	15.251	1,396
9	295.966	698	294.197	690	295.582	16.289	1,388
10	376.985	1,456	374.695	1,438	375.840	19.486	2,894
11	381.684	1,499	377.973	1,470	379.828	20.496	2,989
12	381.452	1,523	382.013	1,527	381.733	19.794	3,050
13	434.646	2.286	435.218	2,292	434.932	22.875	4,578
14	435 228	2,323	436.099	2,332	435.664	22.222	4,655
15	434.938	2,332	435.205	2,335	435.072	22.818	4,667
16	483.086	3,280	486.023	3,320	484.555	24.296	6,600
17	483.730	3,252	485.703	3,279	484.717	24.911	6,531
18	482.876	3,265	486.694	3,317	484.785	24.379	6,582
19	514.179	3,948	513.753	3,941	513.966	25.568	7,889
20	517.982	4,062	524.646	4,167	521.314	26.657	8,229
21	516.257	4,005	519.747	4,059	518.002	25.623	8,064
22	540.679	4,508	536.763	4,443	538.721	26.552	8,951
23	537.897	4,493	538.358	4,501	538.128	27.656	8,994
24	539.291	4,505	536.666	4,461	537.979	26.482	8,966
25	605.222	5,941	606.698	5,970	605.960	29.998	11,911
26	614.502	6,112	610.532	6,035	612.517	28.855	12,145
27	607.320	5,993	608.272	6,012	607.796	30.083	12,005
28	discarded						
29	689.512	7,609	690.182	7,624	689.847	32.111	15,233
30	682,745	7,450	677.588	7,338	680.167	30.334	14,788
31	694.667	7,720	696.466	7,760	695.567	32.218	15,480
32	688.968	7,630	683.204	7,503	686.086	30.610	15,133
33	692.580	7,711	688.481	7,620	690.531	32.169	15,331

DATA FOR FOUR-HOUR RUNS.

Date and time of trial.	Ap.20 pm	Ap.21 pm	Ap.22 am	Ap.21 am	Ap.23 am	Ap.23 pm	Ap.24
Mean draught on trial, ft. and ins.....	8-06¼	8-06$\frac{9}{16}$	8-7$\frac{9}{16}$	8-07⅛	8-06½	8-06$\frac{11}{16}$	8-07
Correspondirg displacement, mean draught, tons	768.	766.	778.	775.8	765.5	767.8	772.
Trim by stern, inches..	3¾	3¼	.....	⅛by he'd	2¼	2⅜	7½
Speed of ship in knots per hour.....	11.908	16.146	16.096	20.250	25.113	27.572	29.783
State of sea.....	smooth	smooth	smooth	smooth	long swell	smooth long swell	smooth long swell
Revolutions of main shafts:							
Starboard	221.9	301.6	299.8	382.9	497.0	551.049	621.609
Port	221.8	300.1	299.4	383.2	496.0	551.190	621.463
Mean revolutions.....	221.85	300.8	299.6	383.1	496.5	551.120	621.536
Slip of propellers, in per cent. of their speed:							
Starboard	14.14	14.34	14.10	15.38	19.15	19.94	23.34
Port	14.10	13.91	13.99	15.45	18.99	19.96	23.32
Mean slip.....	14.12	14.12	14.04	15.41	19.07	19.95	23.33

DATA FOR FOUR-HOUR RUNS—Continued.

Steam pressures:							
Boilers, gage	250.4	249.8	250.1	250.3	262.2	267.	265.
Main steam pipe at throttle	245.5	250.1	243.8	247.2	259.0	260.	250.5
Auxiliary steam pipe..	244.	247.0	240.8	247.5	255.4	256.3	254.
exhaust	2.6	6.7	5.75	4.3	5.7	7.12	10.7
Ahead steam chest, stb'd	200.0	224.0	217.75	216.1	251.8	230.9	237.
Ahead steam chest, port	206.4	226.0	214.0	217.8	248.2	235.2	240.
First stage ahead stbd.	–3.7"	3.21 lbs.	2.75	10.4	42.7	58.9	76.8
First stage ahead port.	–3.7"	3.66 lbs.	3.0	10.4	39.1	56.0	76.4
Nozzles, open	1	{S 1.1 P 1.9	2	4	7	10	13.2
Pressure in lubrication system	19.2	20.	20.	19.3	21.3	20.	22.
Vacuum in condenser, starboard	29.0	29.6	28.5	29.5	28.1	27.8	27.4
port	29.3	29.9	28.5	29.9	28.0	27.7	27.3
Air press. in firerooms, ins. water	1.87	1.0	1.17	3.4	2.3	2.8	3.31
Barometer	29.64	29.74	29.95	29.72	30.04	30.03	29.97
Auxiliaries:							
Revolutions or double strokes:							
Main air pumps, starboard	29.2	32.	28.2	21.8	23.2	24.2	28.8
port	..	..	22.0	26.0	30.2	30.2	32.
circ. pumps, starboard	125.9	119.3	99.6	143.3	148.9	168.2	182.
port	122.5	120.3	112.3	124.0	139.7	168.0	174.
Main feed pumps	..	..	..	..	{S {P24.	23.7 23.3	27.8 27.8
Auxiliary feed pumps, aft	9.3	11.8	12.8	18.0	{S. 2. {P.16.2	2. 20.9	2. 2.
Fuel-oil pumps, forward	..	..	..	..	25.0	25.3	40.1
aft	13.	12.	16.9	25.0	28.1	39.1	46.9
Forced-lubrication pumps	14.	15.	10.	11.5	12.7	14.0	32.
Fire and bilge pumps	..	..	..	..	24.	48.	27.3
Forced-draft blowers.	1062.1	912.	931.	1332.	1503.	1427.	1478.
Dynamos, watts	3182.	3504.	2607.	3612.	3700.	4087.	3012.
Number of boilers in use	1	2	2.	2	4	4	4
Average number of burners in use, each boiler	3	2.85	2.5	4.05	4.	4.65	7
Pressure of oil to burners	88.3	79.8	81.7	144.4	104.	118.8	96.7
Temperatures, Fahrenheit:							
Fuel oil to burners..	105.6	109.0	110.3	120.8	110	96.1	93.45
Injection	37.8	36.5	35.5	36.0	36.2	36.5	38.8
Discharge	56.0	57.5	56.3	62.8	70.0	71.4	77.4
Air-pump discharge...	38.0	38.2	38.1	40.5	49.1	53.8	60.5
Feed	55.	38.2	37.8	67.4	89.8	137.4	120.8
Outside air	44.	40.	38.6	36.	37.9	38.	41.4
Engine-room working level	71.3	68.	73.1	68.0	70.3	74.6	76.1
Fireroom working level	88.1	88.	94.	68.0	77.	69.4	71.3
Horsepower, main shafting:							
Starboard	280.	719.	713.	1494	3595.	4806.	6289.
Port	280.	723.	712.	1500.	3512.	4810.	6284.
Total	560.	1442.	1425.	2994.	7167.	9616.	12573.
Water:							
Pounds, per hour, all machinery	19655	32615	31030	57852	109725	140471	184047
Pounds per hour, per H. P. of main shafts, all machinery	35.098	22.618	21.775	19.323	15.311	14.608	14.638
Fuel oil:							
Analysis and heating power*							
Pounds per hour, all purposes	1489.	1422.	2294.	4653.	8095.	10492	14226
knot run							
Knots, per ton	17.91	14.93	15.72	9.75	6.95	5.89	4.69

DATA FOR FOUR-HOUR RUNS—Continued.

Deduced data:							
Sq. ft. of heat, surface per S.H.P.	8.036	6.241	6.316	3.006	2.512	1.872	1.432
Pounds fuel oil per S.H.P. pr. hr.	2.659	1.680	1.610	1.554	1.129	1.091	1.131
burner pr. hr	496.3	424.9	458.8	574.4	505.9	564.1	508.1
Pounds of water evaporated per pound of fuel oil	13.20	13.47	13.53	12.43	13.55	13.39	12.94
Cooling surface (main condenser per S.H.P)	17.95	6.620	6.699	3.188	1.332	0.993	0.759
Auxiliaries in operation:							
Main air pumps	1	1	1	2	2	2	2
circulating pumps	2	2	2	2	2	2	2
feed pumps.	0	0	0	..	1	1	2
Forced-lubrication pumps	1	1	1	1	1	1	1
Fire and bilge pumps.	2	2	2	2	2	2	2
Auxiliary-feed pumps.	1	1	1	1	..	..	..
Oil-tank pumps	1 oc'ally	1 oc'ally	1 oc'ally	1 oc'ally	1 oc'ally	1 oc'ally	1
Oil-service pumps	1	1	1	1	2	2	2
Forced-draft blowers	1	1	1	{ 3hr 2 { 1hr 1	3	3	4
Dynamos	1	1	1	1	1	1	1
Steering engine	as reqrd	as reqrd	as reqrd	as reqrd	as reqrd	as reqrd	as reqrd
Feed heaters	..	..	..	..	2nd half	1	1
Steam heaters	moder'ly	moder'ly	moder'ly	moder'ly	moder'ly	moder'ly	moder'ly
Steam to galley	as reqrd	as reqrd	as reqrd	as reqrd	as reqrd	as reqrd	as reqrd

*C, 87.00; H, 13.26; S, 0.32; B.T.U. 19,565.

COMPARATIVE PERFORMANCES OF OIL-BURNING TORPEDO-BOAT DESTROYERS ON TRIALS, WITH SOME OBSERVATIONS ON OIL BURNING.

By Captain W. Strother Smith, U. S. Navy, Member.

In answer to the many questions propounded on all trials as to what the other boats of like design had done the accompanying table and these notes are furnished.

To further facilitate the comparison much of the data obtained on trials has been reduced to three speeds, except in the case of the *Trippe*, where six speeds are given. These deduced data are marked with an asterisk in the table. In deducing the figures given, curves were constructed or the rule of cubes used. While this latter method may not give the exact power with accuracy it is sufficiently close for purposes of comparison.

In comparing the powers at different speeds some account must be taken of differences in torsionmeters, state of wind and sea and the efficiency of machinery. It is conceded that the torsionmeters on the *Paulding* and *Drayton* showed slightly higher powers than were required. On the *Perkins* and *Sterett* the powers were measured on one shaft only, because the torsionmeter on the other shaft was known to be out of order. The powers of the other shaft were deduced and are included in the total powers.

The evaporative efficiency of the boilers is not given, because the measurement of the make-up feed was not sufficiently accurate to be used and could not well be measured more accurately. With the exception of the *Trippe* the amounts of oil fuel as given are only approximately correct, but probably more accurate than would be measurements taken in service for runs of the same duration, unless some improved

method of measurement were adopted. In comparing the fuel-oil data some account must be taken of the fact that more care was taken to promote boiler efficiency on some boats than on others. Some were run for speed, regardless of oil consumption, and others, owing to state of weather, required a high power for their sea speeds.

Considering the efficiency of the different burners used none were found to be perfect. Each had some defect, but all had good points. The following points must be observed in burning oil to obtain the best results: (*a*) The oil must be atomized as completely as possible; (*b*) Combustion must commence within an inch or two from the tip of the burner; (*c*) Air must be supplied along the cone of burning oil in increasing quantities and at a sufficient velocity to thoroughly penetrate and mix with the sprayed oil; (*d*) There must be a surrounding volume of air around the burning cone of oil to protect the front boiler casing, the burner-tuyeres, and to ensure complete combustion before the gases leave the combustion chamber.

In forcing oil through the burner for atomization a steady pressure is almost essential. This may be accomplished in several ways; the one usually adopted is to install a large air chamber on the far end of the oil-service piping. Since oil absorbs air very readily a source of supply under pressure must be available at all times to keep up a head. It would be interesting to try a piston accumulator with spring load, spring to be adjustable and used in connection with a long-stroke duplex, or even simplex, pump. The accumulator should be between the burners and the pump.

In atomizing the oil it was previously heated in nearly all cases. This has been proven by experiment, during the above trials and since, to be advantageous. Between 125 degrees and 150 degrees F. seems to be all that is necessary for the best results, but this depends to some extent on the size of the orifice in the burner tip. With the exception of the *Roe* and *Terry* all boats used about the same quality of oil, as to flash point and specific gravity, and of the same degree

of distillation from the crude. It will be seen that all of the trials were conducted with oil showing a higher flash point than required by Government specifications.

It has been noted that with a constant temperature and a wide-open burner the amount of oil burned depends directly upon the pressure at the burner. From this it would seem to be good policy to use a larger number of burners and reduce the pressure to as low a point as will serve to atomize the oil with the burner in use. This will relieve the oil-service pump, require less air pressure for penetration of oil cone and reduce the chances of danger from oil leaks.

The trials of the *Trippe* were conducted under conditions, except weather conditions, exceedingly favorable for the purposes of comparison. In addition to all that had been done on the other boats, given in the table, means of measuring fuel oil were adopted that gave far greater accuracy than on previous trials.

Six four-hour trials were run at about the speeds given in the tables, except the full power. The *Paulding*, *Drayton*, *Perkins*, *Sterett* and *Trippe* were tried on the Rockland course, and the others on the course at Delaware Breakwater. In this connection reference is invited to page 954 of the JOURNAL of the issue of August, 1910.

The *Roe* was tried in July, the *Terry* the latter part of September, the *Paulding* the last of August, the *Drayton* the middle of October, and the *Perkins* the last of September. During these trials the weather varied from intensely hot on the *Roe* to cool and pleasant on boats tried farther north. The remaining boats were tried in cold to very cold water.

Name of boat	*Roe.*			*Terry.*			*Paulding.*			*D*
Type of machinery	Parsons turbines, 3 shafts.			Parsons turbines, 3 shafts			Parsons turbines, 3 shafts.			Parsons tu
Speed, knots	16	25	29.5	16	25	29.5	16	25	29.5	16
* R.P.M., average, main engines	377	632 5	791	382	632.5	793.5	371	615	764	375.5
* Shaft horsepower deduced from trial	1,455	6,600	11,670	1,370	6,786	12,354	1,483	6,950	12,652	1,480
Type of torsionmeter	Chad	burn fla	shlight.	Chad	burn fla	shlight.	Hopk	inson--T	hring.	Hopk
* Water per hour for main engines and auxiliaries, lbs.	37,440	109,250	196,617.5	31,520	112,500	185,407.5	31,360	102,250	162,545	32,480
* Water per S.H.P., main engines only, per hour, lbs.	25.73	16.55	16.848	23.007	16.578	15.008	21.146	14.712	12,847	21.946
* Water per knot, pounds	2,340	4,370	6,665	1,970	4,500	6,285	1,960	4,090	5,510	2,030
* Fuel oil per hour pounds	3,136	8,425	14,750	2,640	8,600	15,458	2,912	8,200	13,363.5	2,528
* Fuel oil per knot, pounds	196	337	500	165	344	524	182	328	453	158
B.T.U. of fuel oil per pound	19,388	19,388	19,388	19,842	19,842	19,430	19,659	19,659	19,659	19,516
Flash point of fuel oil	370	370	370	335	335	375	252	252	252	260
Specific gravity of fuel oil at 60 degrees F	.9082	.9082	.9082	.870	.870	.910	.8825	.8825	.8825	.8785
Total number of burners per boiler	11	11	11	11	11	11	11	11	11	11
System of oil burning	Thornycroft.			Thornycroft.			Normand.			N
Type of boiler	Thornycroft.			Thornycroft.			Normand.			N
Cubic feet of combustion space per boiler	423	423	423	423	423	423	336.11	336.11	336.11	336.11
Number of boilers in use	1	3	4	1	4	4	1	3	4	1
R.p.m. of blowers and number running	1,381..1	880.8..4	1,350....4	1,362..1	1,293..2	1,459....4	1,242..1	1,238..2	1,517...2	1,074..1
Air pressure in firerooms, inches of water	open	1.55	3 265	1.62	1.4	3.4	4.2	4.1	4.87	3.1
Oil pressure at burners and number in use	...11	...14	...44	198...12	191...18	197...44	213...	221...	136.5...	222
Temperature of oil at burners, degrees F	...	...	...	211	204	164	69	74	71.5	60.5
Heating surface of boilers on trial, square feet	4,500	13,500	18,000	4,500	18,000	18,000	4,830.23	14,490.7	19,320.9	4,830.23
* Pounds of oil per hour per sq. foot heating surface	.70	.62	.82	.59	.48	.86	.60	.57	.69	.52
* Pounds of oil per cubic foot furnace volume	7.39	6.64	8.72	6.24	5.08	9.14	8.66	8.13	9.94	7.52
* Water per square foot heating surface per hour	8.32	8.09	10.92	7.00	6.25	10.30	6.49	7.06	8.41	6.72
Average pressure steam at boilers, gage, pounds	241	256.3	239	233	243	250	225	250	252	222
Average temperature feed, degrees F	93.6	104.5	111	94	165	189	222	211	187	230
Displacement, mean, during trial, tons	698	711	711	723	724	722.5	727.5	712	711	727
State of sea	slight swell	smooth	smooth	smooth	smooth	smooth	short, choppy	slight swell	smooth	short, choppy
Force of wind	gentle breeze	gentle	gentle	mod. breeze	gentle to mod. breeze	light airs	mod. breeze	mod. breeze	gentle breeze	fresh
* Shaft horsepowers from standardization curve	1,455	6,600	11,670	1,185	6,335	11,430	1,670	6,925	12,415	1,485
Type of blower, blower engine, and diameter of fan	Sirocco	Terry	30-inch	Sirocco	Terry	30-inch	Bath Ir. Works	Terry	39-inch	Bath Ir Works

* Deduced data.

COMPARATIVE DATA OF TORPEDO-BOAT DESTROYERS.

	Drayton.		*McCall.*			*Burrows.*			*Perkins.*			*Sterett.*			*Warrington.*			
	s turbines, 3 shafts.		Parsons turbines, 3 shafts.			Parsons turbines, 3 shafts.			Curtis turbines, 2 shafts.			Curtis turbines, 2 shafts.			Zoelly turbines, 2 shafts.			
	25	29.5	16	25	29.5	16	25	29.5	16	25	29.5	16	25	29.5	16	25	29.5	12
	622	787.5	367.5	611	763.5	371.5	616	775	289.5	475.1	584.8	293.2	484.5	598.8	268.8	478.2	...	285
	7,078	13,600	1,395	6,789	11,645	1,434	7,014	11,790	1,380	6,304	11,364	1,519	6,592	11,721	1,529	6,996	12,065	639
	nson--Thring.		Föttinger.			Föttinger.			Föttinger.			Föttinger.				Metten.		
	103,000	174,050	32,960	101,500	165,347.5	32,800	103,250	167,707.5	30,400	97,750	164,315	32,560	89,000	174,050	33,680	112,230	176,557.5	20,400
	14.552	12.797	23.627	14.951	14.199	22.872	14.721	14.225	22.029	15.506	14.458	21.435	13.501	14.845	22.027	16.045	14,634	31.925
	4,120	5,900	2,060	4,060	5,605	2,050	4,130	5,685	1,900	3,910	5,570	2,035	3,560	5,900	2,105	4,495	5,985	1,700
	7,800	15,428.5	2,640	7,400	13,363.5	2,592	8,350	13,039	2,528	8,100	15,753	2,736	8,600	16,933	3,408	9,625	165,790	1,596
	312	523	165	316	453	162	334	442	158	324	534	171	344	574	213	385	562	133
	19,516	19,516	19,618	19,618	19,618	19,598	19,598	19,598	19,752	19,752	19,752	19,679	19,679	19,679	19,510	19,510	19,510	19,607
	260	260	300	300	300	273	273	273	275	275	275	260	260	260	270	270	270	250
	.8785	.8785	.8865	.8865	.8865	.8958	.8958	.8958	.881	.881	.881	.8805	.8805	.8805	.8961	.8961	.8961	.877
	11	11	11	11	11	11	11	11	7	7	7	7	7	7	11	11	11	11
	ormand.		Schütte--Koerting.			Schütte--Koerting.			Fore River.			Fore River.			†Schütte--Koerting.			
	ormand.		Thornycroft.			Thornycroft.			Yarrow.			Yarrow.			White-Forster.			
	336.11	336.11	437	437	437	437	437	437	525	525	525	525	525	525	394.74	394.74	394.74	336.11
	3	4	1	3	4	1	4	4	2	3	4	2	4	4	2	4	4	1
	1,200..2	1,573...2	1,426..1	1,185..3	1,391...4	1,388..1	1,061..4	1,441...4	1,372..1	1,330..3	1,413...4	1,222..1	1,515..2	1,537...4	714...1	1,184..2	1,212...4	735...1
	3.5	4.83	3.2	3.55	4 63	4.45	4.80	4.95	1.17	2.07	2.45	1.41	2.75	3.28	1.38	3.2	4.85	1.6
	260...14	219...37	121...7	120...25	148...41	136..6.6	105..25.5	127.5..43	69.5..6	85.5..15	100...28	158...4	112...16	115...28	204...7	201...24	197...37	191...
	59	61	218	206	220	217	194	223	...	...	149.5	138	111	128	165	165	171	209
	14,490.7	19,320.9	4.800	14,400	19,200	4,800	19,200	19,200	9,000	13,500	18,000	9,000	18,000	18,000	9,000	18,000	18,000	4,830
	.54	.80	.55	.51	.70	.54	.44	.68	.28	.60	.86	.30	.48	.94	.38	.54	.92	.3
	7.74	11.48	6.04	5.64	7.65	5.93	4.78	7.46	2.41	5.14	7.50	2.61	4.10	8.06	4.31	6.09	10.50	4.7
	7.11	9.00	6.87	7.05	8.61	6.83	5.37	8.73	3.38	7.24	9.13	3.62	4.94	9.67	3.74	6.21	9.81	4.2
	255	254	240	252	249	254	260	253	258	256	257	265	263	263	248.5	246	246	177
	219	204	216	184	165.9	216	184.3	169.4	64	66	73	55.5	67	80	190	135	140	221
	72	721	738.6	731	738	735.2	720	720.5	775.5	757	765	765	756.5	754	726.8	735	729.2	739
	smooth	longswell to smooth	smooth short chop	smooth	mod. short following	smooth short chop	short mod.	slight swell	smooth to slight swell	smooth to slight swell	smooth	slight swell	slight swell	short and rough	smooth	light to mod.	slight swell	smoot
	light to mod.	gentle breeze	mod. to fresh	gen'le breezes	mod. to fresh	fresh	gentle breeze	mod.	light to mod.	mod. breeze	gentle to fresh breeze	mod. breeze	light to mod. breeze	mod. to fresh breeze	strong breeze	light to mod.	light to gentle	mod.
	7,120	13,440	1,300	6,400	11,425	1,305	6,500	11,950	1,295	6,200	11,000	1,345	6,190	11,210	1,460	6,720	11,940	620
	Terry	39 inch	Sturtevant	Terry	36-inch	Sturtevant	Terry	36-inch	Sirocco	Kerr	27-inch	Sirocco	Kerr	27-inch	Sturtevant	Terry	39-inch	

† B. & W. air distributer attache

ARATIVE DATA OF TORPEDO-BOAT DESTROYERS.

		Burrows.			*Parkins.*			*Sterett.*			*Warrington.*			*Trippe.*			
		Parsons turbines, 3 shafts.			Curtis turbines, 2 shafts.			Curtis turbines, 2 shafts.			Zoelly turbines, 2 shafts.			Parsons turbines, 3			
	29.5	16	25	29.5	16	25	29.5	16	25	29.5	16	25	29.5	12	16	20	2
	763.5	371.5	616	775	289.5	475.1	584.8	293.2	484.5	598.8	268.8	478.2	...	285	380.5	485	63
	11,645	1,434	7,014	11,790	1,380	6,304	11,364	1,519	6,592	11,721	1,529	6,996	12,065	639	1,504	3,006	7,24
r.		F	ötting	er.	F	ötting	er.	F	ötting	er.		Metten.			Hopk	inson–Th	ring to
	165,347.5	32,800	103,250	167,707.5	30,400	97,750	164,315	32,560	89,000	174,050	33,680	112,250	176,557.5	20,400	35,280	53,500	107,50
	14.199	22.872	14.721	14.225	22.029	15.506	14.458	21.435	13.501	14.845	22.027	16.045	14,634	31.925	23.457	17.797	14.837
	5,605	2,050	4,130	5,685	1,900	3,910	5,570	2,035	3,560	5,900	2,105	4,495	5,985	1,700	2,205	2,675	4,30
	13,363.5	2,592	8,350	13,039	2,528	8,100	15,753	2,736	8,600	16,933	3,408	9,625	165,790	1,596	2,784	4,060	8,67
	453	162	334	442	158	324	534	171	344	574	213	385	562	133	174	203	34
	19,618	19,598	19,598	19,598	19,752	19,752	19,752	19,679	19,679	19,679	19,510	19,510	19,510	19,607	19,607	19,607	19,60
	300	273	273	273	273	275	275	260	260	260	270	270	270	250	250	250	25
	.8865	.8958	.8958	.8958	.881	.881	.881	.8805	.8805	.8805	.8961	.8961	.8961	.877	.877	.877	.877
	11	11	11	11	7	7	7	7	7	7	11	11	11	11	11	11	1
e	rting.	Schü	tte–Koe	rting.	F	ore Riv	er.	F	ore Riv	er.	†Schü	tte–Koe	rting.			No	rmand.
r	oft.	T	hornycr	oft.		Yarrow.			Yarrow.		Wh	ite-Fors	ter.			No	rmand.
7	437	437	437	437	525	525	525	525	525	525	394.74	394.74	394.74	336.11	336.11	336.11	336.11
3	4	1	4	4	2	3	4	2	4	4	2	4	4	1	1	2	3
3	1,391...4	1,388..1	1,061..4	1,441...4	1,372..1	1,330..3	1,413...4	1,222..1	1,515..2	1,537...4	714...1	1,184..2	1,212...4	735...1	1,083..1	898...1	1,137 ..
	4 63	4.45	4.80	4.95	1.17	2.07	2.45	1.41	2.75	3.28	1.38	3.2	4.85	1.6	2.9	1.9	2.65
5	148...41	136..6.6	105..25.5	127.5–43	69.5..6	85.5..15	100...28	158..4	112...16	115...28	204...7	201...24	197...37	191...3	291...7	286...9	280...2
6	220	217	194	223	...	...	149.5	138	111	128	165	165	171	209	197	207	200
o	19,200	4,800	19,200	19,200	9,000	13,500	18,000	9,000	18,000	18,000	9,000	18,000	18,000	4,830.2	4,830.2	9,660.5	14,490.
	.70	.54	.44	.68	.28	.60	.86	.30	.48	.94	.38	.54	.92	.33	.58	.42	.60
	7.65	5.93	4.78	7.46	2.41	5.14	7.50	2.62	4.10	8.06	4.31	6.09	10.50	4.75	8.28	6.02	8.60
	8.61	6.83	5.37	8.73	3.38	7.24	9.13	3.62	4.94	9.67	3.74	6.21	9.81	4.22	7.30	5.54	7.42
2	249	254	260	253	258	236	257	265	263	263	248.5	246	246	177	226	244	259
4	165.9	216	184.3	169.4	64	66	73	55.5	67	80	190	135	140	221	207	201	206
1	738	735.2	720	720.5	775.5	757	765	765	756.5	754	746.8	735	729.2	739	726	725	730
h	mod. short following	smooth short chop	short mod.	slight swell	smooth to light chop	smooth to slight swell	smooth	slight swell	slight swell	short and rough mod.	smooth	light to mod.	slight swell	smooth	short swell	short chop	short chop
e	mod. to fresh	fresh	gentle breeze	mod.	light to mod.	mod. breeze	gentle to fresh breeze	mod. breeze	light to mod. breeze	to fresh breeze	strong breeze	light to mod.	light to gentle	mod.	light to gentle breeze	mod. to fresh breeze	fresh t very fre wind
o	11,425	1,305	6,500	11,950	1,295	6,200	11,000	1,345	6,190	11,210	1,460	6,720	11,940	620	1,440	3,010	7,07
y	36-inch	Sturte-vant	Terry	36-inch	Sirocco	Kerr	27-inch	Sirocco	Kerr	27-inch	Sturte-vant	Terry	39-inch	B	ath Iron	Works,	Terry t

† B. & W. air distributer attached to S.-K. system.

			Ammen.						*Walke.*					
, 3 shafts.			Parsons turbines, 3 shafts.						Curtis turbines, 2 shafts.					
25	27.5	29.5	12	16	20	25	27.5	29.5	12	16	20	25	27.5	29.5
634	719.5	806	277	373.5	470.5	615.5	694	779	223 3	297.7	377.6	494.0	549.2	611.2
,245	10,122	12,770	675	1,490	2,860	6,989	9,173	12,677	573	1,400	2,884	7,071	9,541	12,218
torsi	onmeter.				Föttinger.						Föttinger.			
,500	146,712.5	180,392.5	19,980	33,200	48,800	93,600	142,900	178,300	19,800	30,600	55,700	108,300	139,500	176,800
837	14.494	14.126	29.600	22.282	17.063	13.392	15.578	14.065	34.555	21.857	19.313	15.316	14.621	14.470
,300	5,335	6,115	1,665	2,075	2,440	3,745	5,195	6,045	1,690	1,912	2,785	4,332	5,070	5,995
,675	11,440	14,573	1,880	3,055	4,450	8,345	12,630	15,230	1,495	2,255	4,490	8,005	10,395	13,650
347	416	494	157	191	222.5	334	459	516	125	141	224	320	378	463
,607	19,607	19,607	19,503	19,503	19,503	19,503	19,503	19,503	19,565	19,565	19,565	19,565	19,565	19,565
250	250	250	288	288	288	288	288	288	260	260	260	260	260	260
77	.877	.877	.8935	.8935	.8935	.8935	.8935	.8935	.8795	.8795	.8795	.8795	.8795	.8795
11	11	11	11	11	11	11	11	11	7	7	7	7	7	7
nd.				Schütte-Thorn-		Koertingycroft.				Fore Fore		River. River.		
5.11	336.11	336.11	437	437	437	437	437	437	525	525	525	525	525	525
	4	4	1	1	2	3	4	4	1	2	2	4	4	4
7 ..2	1,263...2	1,462...2	1,035..1	1,244..1	1,024..2	1,098..3	1,134..4	1,293..4	1,062..1	931..1	1,332..2	1,003..2	1,427..3	1,478..4
2.65	2.8	4.1	3 58	4.43	4.09	3.9	5.0	5.84	1.87	1.17	3.4	2.3	2.8	3.31
...21	237-29.5	280...40	125...5	137...8	117..11	104.5..27	116..35	128.44	88..3	81.7..5	144..8	109..16	118.8..18	96.6..28
	196	196	206	219	215	221	220	222	106	110	120	110	96	93.5
90.7	19,320 9	19,320.9	4,800	4,800	9,600	14,400	19,200	19,200	4,500	9,000	9,000	18,000	18,000	18,000
.60	.59	.75	.392	.632	.463	579	.657	.793	.332	.255	.499	.444	.577	.738
8.60	8.51	10.84	4.30	6.97	5.09	6.36	6.83	8.24	2.85	2.15	4.28	3.81	4.95	6.50
7.42	7.59	9.34	4.2	6.7	5.1	6.5	7.44	9.28	4.4	3.4	6 2	6.0	7.75	8.22
	226	250.5	242	240	260	252	250	250	250	250	250	262	267	265
5	199	179	70	72	85	62.5	80	163	55	38	67	90	137.4	120.8
	722	733	747.5	737.7	737.4	730.3	738.0	735.9	768	778	773.8	765.5	767 8	772
ort	short		long	short	old swell	long	old	long				long	smooth	smooth
hop	chop	smooth	high swell	chop	& short chop	swell	swell	high swell	smooth	smooth	smooth	swell	long swell	long swell
h to fresh ind	gentle to mod.	mod.	gentle breeze	fresh breeze	mod. to fresh breeze	gentle breeze	gentle to mod. breeze	mod. breeze	gentle breeze	gentle breeze	fresh breeze	mod. breeze	gentle breeze	light breeze
,075	10,060	12,800	475	1,310	2,700	6,640	9,265	12,105	600	1,370	2,910	6,970	9,460	12,130
y tur	bine, 39-i	nch.		Sturtevant	Terry	36-inch				Sirocco	Kerr	27-inch		

NOTES ON THE CURTIS TURBINE.

By Lieutenant O. L. Cox, U. S. N., Member.
Graduate of the School of Marine Engineering.

The Curtis turbine is a compound impulse turbine, the compounding being both for pressure and velocity.

In an impulse turbine, the internal energy of the steam is first changed into velocity, by expansion in a suitable nozzle, and the jet of steam directed against the moving blades. In passing through the moving blades, the steam gives up part of its velocity to the blades, this part of the velocity being changed into mechanical work.

The simplest possible impulse turbine is the DeLaval, which consists of a nozzle or row of nozzles, and a single row of moving blades. The steam is expanded, from the initial pressure to that of the condenser, in the nozzle.

The maximum efficiency of a single-wheel impulse turbine is when the blades move with a velocity that is one-half the velocity of the jet of steam, and as steam velocities of nearly 4,000 feet per second are attained when high-pressure steam is expanded into a vacuum, it is seen that for maximum efficiency the blades must move at a velocity of 2,000 feet per second. Such a speed is prohibitive for marine propulsion, and some means must be taken to reduce it.

This is done by compounding the pressure drop, that is, by using several sets of nozzles, thereby reducing the drop in each set, and reducing the exit velocity of the steam accordingly.

Even by doing this the turbine would be too long for use if there were only one row of blades in each stage to extract the velocity from the steam.

Therefore the turbine is compounded for velocity also, that is, instead of only one row of moving blades per wheel there are several rows. Each row extracts approximately twice its own velocity from the steam, and there are 2, 3 or 4 rows of moving blades used, depending on the relative velocities of the steam and the blades.

If the nozzle were placed exactly in the plane of the wheel, and the blades were of such shape as to completely reverse the direction of the steam, the blades would extract exactly twice their own velocity.

This condition is not possible, owing to practical reasons, the nozzles being placed at an angle with the wheel, and the blades not giving a complete reversal of direction of the steam.

Turbines are inherently high-speed machines, and for this reason are not suitable for slow ships without some sort of reduction gear. A turbine designed to give as good economy at low speeds as the reciprocating engine would be of prohibitive weight, and at a high number of revolutions the propeller efficiency drops off rapidly. Up to speeds of 15 or 16 knots, the turbine is less economical than a well designed reciprocating engine, while at high speeds the turbine runs with about equal economy.

CONSTRUCTION.

The general construction of the turbine is shown in the plate, which is a longitudinal section of one of the turbines on the U. S. S. *North Dakota,* and represents one of the latest types of Curtis turbines in our service.

The turbine consists of two general parts—the stator, or stationary parts, and the rotor, or rotating parts.

The stator includes the casing, steam chests, diaphragms and fixed blades. The casing is made of cast steel, and is divided into sections so as to be easily lifted.

The rotor consists of the following: A hollow-steel shaft, the forward end of which forms the thrust shaft, and the after end is attached to the line shaft, the after end being the largest; cast-steel hubs are forced on the shaft by hydraulic

pressure and then keyed. Bolted or riveted to the hubs are discs of rolled-steel plate (cast-steel in some turbines). The rims to which the moving blades are attached are bolted or riveted to the discs.

The turbine is divided into "stages" by means of diaphragms which are secured to the casing of the turbine. This principle is easily understood if we consider that the turbine is made up of several single-wheel turbines in series. The first turbine working from an initial steam pressure of say 280 pounds absolute to 75 pounds absolute, the second turbine

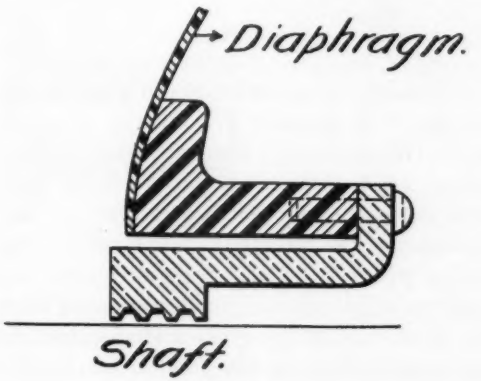


Fig. 1.

working from 75 pounds to 40 pounds, and so on throughout the series of turbines. Remembering that each turbine is called a "stage." These diaphragms are made of steel plate (cast steel in some turbines) and have a special bushing around the shaft to prevent steam leakage. A sketch of this bushing is shown in Fig. 1. It is made of bronze, with V-shaped grooves cut around it, and is so constructed that, in case of any unusual expansion of the shaft or diaphragm, the bushing will spring a little. Besides the wear will come on the small soft points of the grooves, which will wear down easily.

The leakage, through a well constructed bushing, should be less than 5 per cent. The steam leaks directly into the

next stage without doing any useful work, similar to steam leaking past the piston of a reciprocating engine.

The latest type of Curtis turbine installed (H. M. S. *Bristol*) is a compound turbine. That is, it is divided into a high and low-pressure turbine on the same shaft, with a bearing between. That is to prevent having to use a long unsupported shaft. It increases the weight somewhat, but is a more economical turbine, as the number of stages have been increased thus reducing the pressure drop in each stage and thereby increasing the efficiency.

NOZZLES.

Steam expanding in a nozzle from a given initial pressure P' to a given back pressure P'', falls to a pressure in the throat of the nozzle which is about .58 of the initial pressure, regardless of the back pressure, provided the back pressure is not more than .58 of the initial pressure.

The diameter of the turbine is quite large, so that for considerations of strength the steam pressure in the first stage is limited, otherwise an exceptionally heavy casing must be used; also there should not be too great a temperature range from the high-pressure end to the low-pressure end of the turbine.

Pressures of not much over 100 pounds per square inch being desirable in the turbine, and as boiler pressures of 300 pounds are common, it becomes necessary to use a nozzle which will expand the steam to a lower pressure than .58 of the initial pressure. Such a nozzle is called an expansion nozzle, and is shown in Fig. 2. The amount the nozzle itself enlarges depends on the final pressure desired. A nozzle which has the smallest section at the exit or mouth is called a non-expanding nozzle, and is the type used between the intermediate stages of the turbine. As used in the turbine this nozzle is practically of the same length as the expanding nozzle, but the walls are parallel, so that the area of the mouth is the same as at the throat.

Fig. 3 shows a non-expanding nozzle, the pressure at exit

being .58 of the initial pressure, provided the back pressure is not higher.

The nozzles for the intermediate stages are carried in the diaphragms between the stages.

Steam expanding in a nozzle expands adiabatically, that is, without the addition or extraction of heat as heat, the heat given up being changed into velocity. The amount of heat converted into velocity is called the "heat drop" in the nozzle.

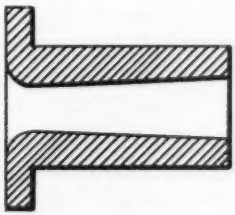


Fig. 2.

Fig. 3.

Let P' be 280 pounds absolute with 50 degrees F. superheat.
And P'' be 100 pounds absolute.
From a total heat-entropy chart we get the following data:
H'=1,238.4 B.T.U. (Total heat in initial steam).
H''=1,151.5 B.T.U. (Total heat in the steam after adiabatic expansion to 100 pounds absolute).
$H'-H''$=86.9 B.T.U. (Heat drop in the nozzle.)
To find the theoretical exit velocity of the steam, use the formula:— $V^2=2gh$.

The kinetic energy of the steam=$K.E.=\frac{1}{2}mV^2=\frac{WV^2}{2g}$.

All heat given up is changed into kinetic energy

$$\therefore K.E.=\frac{WV^2}{2g}=h \text{ foot pounds.}$$

For 1 pound of steam $W=1$, and $V^2=2gh$, where
V=Theoretical velocity of the steam.

g=Gravitational constant (32.2).

h=Heat drop in the nozzles expressed in foot pounds.

$=(H'—H'')\times 778$. (778 ft.-lbs. equals 1 B.T.U.)

or $V=224\sqrt{H}$, where H is the heat drop in the nozzles in B.T.U.'s.

Substituting in the above example, wet get for the theoretical exit velocity of the steam, 2,086 feet per second. Owing to friction of the steam on the walls of the nozzle, the actual exit velocity is less than this.

The smaller the angle between the plane of the nozzle and the plane of the wheel the more efficient is the action of the steam on the blades. For practical reasons this angle is made exactly 20 degrees for all except the last stage nozzles, where it is increased to 25 degrees.

The nozzles are arranged in an arc of a circle and discharge directly against the first row of moving blades. When the turbine is designed, the approximate water rate is determined (pounds of water per horsepower per hour). Then, knowing the horsepower to be developed, the designer knows how much steam must flow through the turbine in a given time, and he is able to calculate the total nozzle area to deliver this amount of steam. The cross section of all nozzles throughout their length is rectangular, with slightly rounded corners. At the throats this cross section is a square with rounded corners. The radial height of the nozzles is kept practically constant throughout their length, so that all enlargement of the nozzle itself is lateral, the shape of the opening gradually changing from the square cross section at the throat to a long rectangle at the mouth. The total area calculated as above is increased about 30 per cent. to allow for overload.

Blades of about one and one-half inches in height are used for the first row of moving blades, and this limits the radial height of the nozzle.

The mouths of the nozzles form practically a continuous band, with just the necessary thickness of walls between. (See Barton, page 409.) This is to give a continuous band of steam at exit.

If the steam did not issue in a practically solid band there would be large losses from eddy currents. When nozzles are opened care should be taken that adjacent nozzles are opened, even then there is a considerable loss at the two end nozzles.

The nozzles for the first-stage cover an arc of about 45 degrees, and as the steam expands from stage to stage its specific volume increases. It being necessary to limit the height of the first row of moving blades, this extra volume must be allowed for by increasing the number of nozzles from stage to stage, until they completely fill the circumference of the diaphragm (about the 5th stage). From there on the size of the nozzle is increased, consequently the length of the blades.

The steam in going through a stage does not spread very much, but goes through as a band of steam, just a little wider than the arc of the nozzles. From this it is seen that it is not necessary to have the intermediate fixed blades all around the periphery, except when the nozzles extend all the way around. The stationary blades extend only a little beyond the arc occupied by the nozzles.

When running, the stage is filled with steam of the same pressure throughout, but the working steam passes through in a direct path.

The only expansion of steam, in this type of turbine, is in the nozzles, therefore there is no tendency for the steam to leak over the tips of the blades as in the Parsons' type, where there is a drop in pressure from one row to the next.

To insure a constant pressure throughout a stage, holes are drilled through the disc wheels.

BLADES.

The blades are usually made of bronze, which has been extruded through a die of proper shape. The metal comes out of the die in a long bar, which is then cut into pieces of the proper length for the blades, and the ends milled to shape.

The blades are then assembled, by sections, in a steel channel base, and the end of the blade riveted over. The chan-

nel base is sawed out to permit of flexibility, as it must then be bent to fit the rim.

After the section is assembled on the rim the channel base is caulked in place, and a shroud ring put over the ends of the blades and the ends riveted over. The shroud ring serves to keep the blades in their proper position and also makes a more rigid construction.

Samples of Curtis blades are in the model room, and show the method of securing the blades in the channel base and shroud ring, also the method of sawing out the channel base.

The rows of moving blades, in any stage, are far enough apart to allow a row of stationary or intermediate blades to be inserted between. These stationary blades are mounted similarly to the moving blades and are secured to the casing of the turbine, their purpose being to reverse the direction of the steam and to direct it on the next row of moving blades at the proper angle.

The blades in any particular row are exactly alike, but differ from row to row throughout the stage. The same quantity of steam flows through each row in the same time, and as the velocity of the steam decreases from row to row throughout the stage, the area for steam passage must be greater. This is done by making the blades longer, also by increasing the pitch (distance between adjacent blades in the same row).

Fig. 5 shows the change in shape of the blade section from row to row. The blades becoming thinner and the angles larger, the stationary blades being curved in the opposite direction from the moving blades. Also see Fig. in Barton, page 409, which shows the arrangement of the blades and nozzles of a Curtis turbine.

Owing to the large pressure drop necessary for the first stage it requires four rows of moving blades to extract the velocity, the other stages having three rows of moving blades, there being one row less of fixed than of moving blades.

The object sought by the designer is to so design the turbine that the absolute exit velocity of the steam from the last row of moving blades in any stage shall be as low as

practicable, and that its direction shall be parallel to the shaft. A certain amount of residual velocity is necessary, so that the steam will get clear of the blades.

ACTION OF THE STEAM IN THE TURBINE.

The action of the steam in the turbine is as follows:

Steam enters the steam chest at a little below boiler pressure; from the steam chest it is expanded through a set of nozzles, reducing the pressure and thereby gaining velocity, the amount of this velocity depending on the heat drop in the nozzle. The steam strikes the first row of moving blades at the angle at which the nozzle is placed, passes through the first row of blades, giving up some of its velocity, and issues in a jet against the first row of fixed blades. It passes through the fixed blades, where its direction is changed, and is directed against the second row of moving blades at the proper angle, and so on throughout the turbine.

Fig. 4 is a theoretical velocity diagram, showing the action of the steam on the moving blade. The only difference between this and an actual diagram is, that in the actual diagram allowance must be made for the friction of the steam on the blades, both moving and fixed.

BD is the plane of the wheel carrying the blade.

AB is the plane of the axis of the nozzle, and to scale, represents the direction and velocity of the steam as it issues from the nozzle, and makes an angle of 20 degrees with BD.

BC to the same scale represents the peripheral velocity of the blade.

Then by compounding these two velocities we get AC as the relative entrance velocity of the steam. The angle ACD is the angle of the *back* of the blade, so that the steam may enter without shock.

If it were not for friction the steam would leave the blade at the same relative velocity as it entered. In this example friction has been disregarded, and also the exit angle DCE has been made the same as the entrance angle, though in practice it is usually made smaller.

Then we have CE for the relative exit velocity equal to AC the relative entrance velocity.

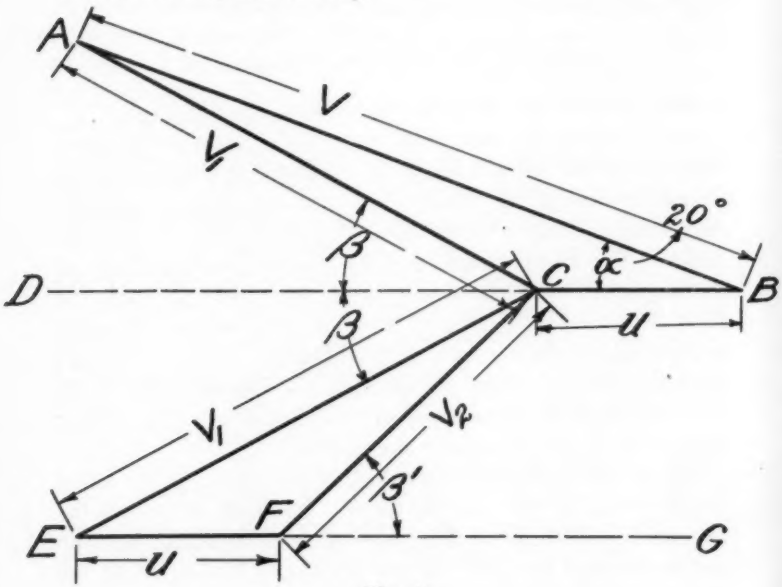


Fig. 4.

To find the absolute exit velocity, the relative velocity must again be compounded with the blade speed (reversed this time). This gives CF as the direction and velocity of the exit steam.

Let V=Absolute entrance velocity.

V_1=Relative entrance velocity and relative exit velocity.

V_2=Absolute exit velocity.

The kinetic energy of the entering steam per pound is:—

$$\tfrac{1}{2}mV^2 = \frac{WV^2}{2g} = \frac{V^2}{2g}, \ (W = 1).$$

The total energy extracted by the blade shown is:—

$$\left\{\frac{V^2}{2g} - \frac{V_1^2}{2g}\right\} + \left\{\frac{V_1^2}{2g} - \frac{V_2^2}{2g}\right\} = \frac{V^2}{2g} - \frac{V_2^2}{2g} = \text{ft.-lbs. per lb. of steam.}$$

Fig. 5 shows a theoretical velocity diagram for a stage having three rows of moving blades and two rows of fixed blades. It shows the change in the shape of the blade from row to row, also the increasing angles of entrance and exit.

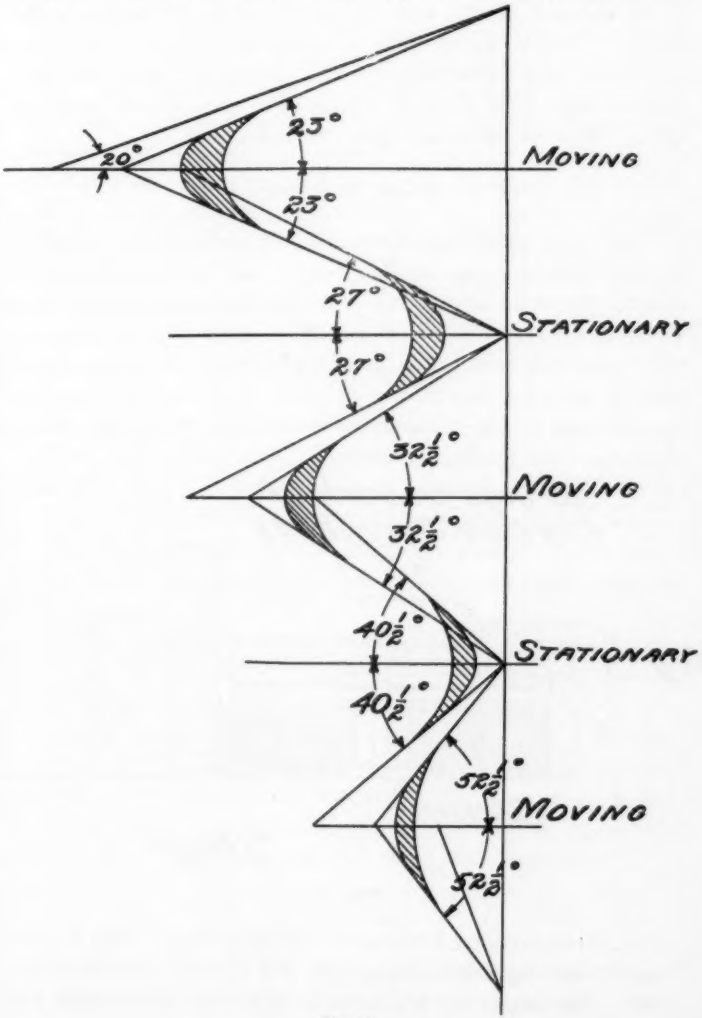


Fig. 5.

There being no restricted steam ports, and the exhaust pipe leading directly from the last row of moving blades to the condenser, the turbine can utilize a much higher vacuum than the reciprocating engine.

An increase of one inch in vacuum from 27 inches to 28 inches results in an increase in economy of from 6 to 8 per cent., exclusive of the extra power for the air pump, 29-inch vacuum and over is quite common in commercial practice, though 28.5 inches is very good on board ship.

STUFFING BOXES.

Where the shaft passes through the ends of the casing it is provided with carbon stuffing boxes, which prevent the steam leaking out at the ahead end, or air leaking in at the back end where a vacuum exists. Steam of 3 or 4 pounds pressure above atmosphere is led to the space between the carbon rings, thereby insuring that no air will leak in at the low-pressure end, causing a loss of vacuum, and allowing that much more air to get into the feed water.

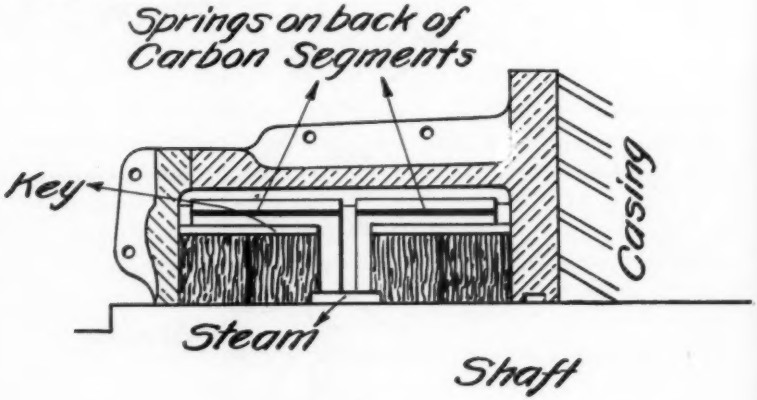


Fig. 6.

Fig. 6 is a section of the carbon packing rings. The carbon is in circular segments with a little end clearance to allow for wear. The segments are held in place by a key, and are

pressed against the shaft by springs, the joints of the two adjacent rings being broken.

DRAINS.

Drains lead from the carbon stuffing boxes to one of the low-pressure stages. An external drain pipe, fitted with a valve, leads from a drain pocket in each stage, to the next lower stage, the exhaust chamber being drained to the main condenser by means of an ejector.

These drains serve to clear the turbine of water, but should be kept closed except when actually in use. They should not be used when underway except when absolutely necessary, as they permit the steam to flow directly through the turbine without doing any useful work.

CLEARANCES.

The thrust bearing being at the forward end of the turbine, and thus fixing the relative positions of casing and rotor, the axial clearance is least at this end, being about one-tenth of an inch. This increases towards the low-pressure end, where, in very long turbines, it is as much as one-half of an inch. as the difference of expansions of casing and rotor must be allowed for.

The axial clearance should be as small as possible, though the radial clearance does not make much difference, as there is no difference in pressure throughout the turbine stage.

A clearance indicator is fitted on both ends of the turbine, by means of which the clearance between the moving and stationary parts can be ascertained. The clearance should always be taken and recorded before starting to warm up, when ready to get underway, and when well underway.

GOVERNING.

Governing is done both by the nozzles and the throttle valve. The proper number of nozzles are opened to assure the desired

number of revolutions, and the final adjustment is made with the throttle valve. Care should be taken to see that all nozzles in use are fully open.

BALANCING.

When the rotor is assembled it is very carefully balanced on knife edges, as any unbalanced weight at the high speed of revolution might cause considerable damage, even wrecking the turbine.

To partly balance the propeller thrust, the last few stages (6th, 7th, 8th and 9th of the *North Dakota*) are mounted on a drum instead of on wheels. The steam pressure acting on the forward face of the drum, counterbalances the thrust of the propeller and the exhaust pressure on the after face of the drum. Theoretically this totally balances at just the designed speed, but it nearly balances at all speeds, so that only small thrust block is necessary, the principal object of which is to hold the turbine rotor in accurate adjustment, and to take the thrust when reversing.

REVERSING.

The turbine is a unidirectional machine, therefore it necessitates the use of another turbine for reversing. The reversing turbine is mounted on the same shaft as the main turbine, and has the same exhaust chamber.

On the *North Dakota* there are twenty ahead nozzles, four of which are always open, the other sixteen being independently operated by screw-down valves.

For reversing there are sixteen nozzles, ten of which are always open.

When the signal comes to reverse, it is necessary to shut off the ahead throttle and then open the astern throttle, but no damage would result if one of the throttle valves was not entirely closed when the other was opened; the turbine would revolve in the direction in which the most power was exerted.

The losses in the turbine may be divided as follows:

1. Steam friction in the nozzles.
2. Steam friction in the blades.
3. Resistance of steam on revolving surfaces.
4. Loss due to absolute exit velocity of the steam from the last row of moving blades in each stage.
5. Radiation.
6. Bearing losses.
7. Losses between nozzles and moving blades, and between the fixed and moving blades.
8. Leakage through the diaphragm bushings.
9. Loss due to windage, reverse turbine revolving in a partial vacuum.

Advantages Claimed for the Curtis Over the Parsons Turbine.

1. About 1-40th the number of blades that are required in the Parsons.
2. Blades are heavier and fixed in a more substantial manner.
3. Large radial clearances permissible.
4. Fewer number of turbines required for the same economy.
5. Higher steam pressures and superheat are permissible, as the greatest pressure in the casing is only ⅓ the boiler pressure, thus permitting the use of large diameters and lower revolutions.
6. No dummy pistons necessary, thus reducing the leakage and allowing larger clearances.
7. Lower revolutions, thus permitting the use of larger propellers, thereby increasing the propeller efficiency.
8. No danger from reversing instantly, owing to the large clearances.
9. Better control of the steam entering the turbine.
10. No danger from priming. The blades are large and secure so that water will not injure them, the turbine slowing down in case the boilers prime.

CARE AND OPERATION.

The turbine should be warmed up very carefully, from three to four hours being required, turning the turbine a quarter of a turn every ten or fifteen minutes with the turning engine. It is better to start the Curtis turbine cold than only partially warmed up. When getting underway the speed should be worked up gradually.

In warming up, steam should be admitted at both ends of the turbine. Start all auxiliaries the same as for reciprocating engines.

Turn steam on carbon packing glands. Flood the bearings.

Open all drains and clear turbine of water; carefully examine forced-lubrication system and see that everything works properly.

When steam is on main line, crack the by-passes of the ahead and astern operating valves so as to warm up piping and steam chests.

About one-half hour before getting underway disconnect turning gear and see everything clear.

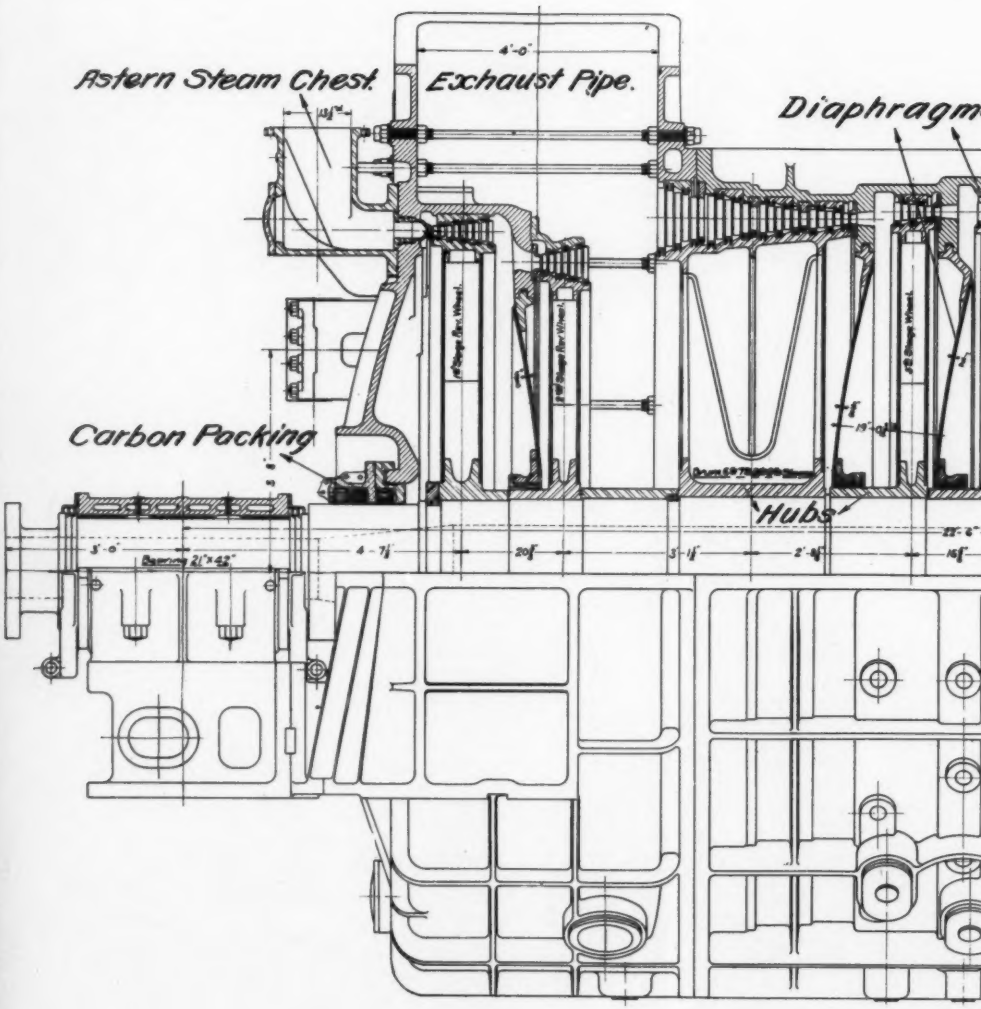
In trying the turbine, open steam to the ahead end, just enough to move it several revolutions, shut off ahead and do the same with the reverse end.

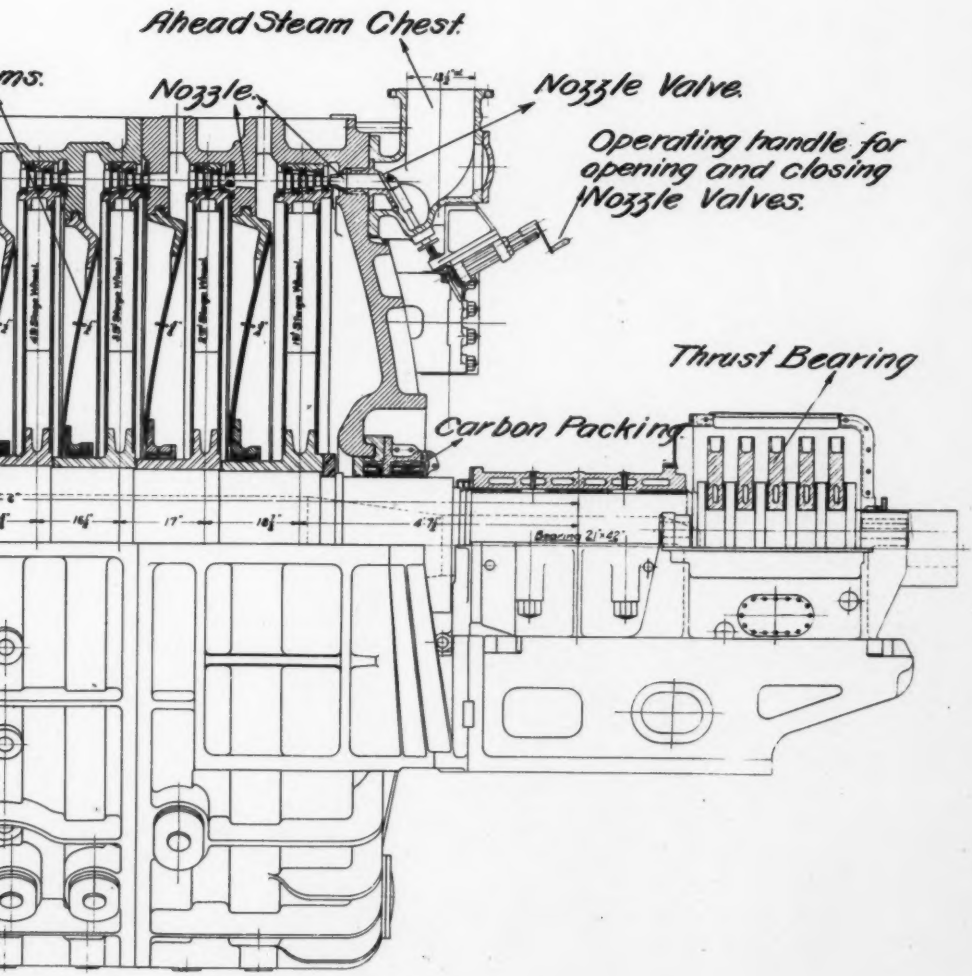
Take clearance before warming up, and when everything is ready to get underway and when going ahead at full speed. When underway, open enough nozzles to give the required speed and make small changes with the throttle valve. Nozzles should be open all the way and not partly open.

To increase the economy turn the auxiliary exhaust into one of the low-pressure stages according to the pressure.

When securing, care should be taken to see that the turbine is free from water before the air pump is secured.

Owing to relatively small clearances in the turbine great care must be exercised in all adjustments.


Astern Steam Chest.
Exchaust Pipe.
Diaphragm
Carbon Packing
Hubs
4'-0"
5'-0"
4 - 7½
20⅝
3'-1½
2'-8½
16⅝


Ahead Steam Chest.
Nozzle.
Nozzle Valve.
Operating handle for opening and closing Nozzle Valves.
Thrust Bearing
Carbon Packing

CURTIS TURBINE DESIGN.

By Lieutenant Ormond L. Cox, U. S. N., Member.
Graduate of the School of Marine Engineering.

The Curtis turbine is of the compound impulse type, and it is necessary in taking up the study of its theory to first consider the action of a jet of moving steam on a fixed plane, and afterwards on a moving plane.

In the theoretical study, the planes are all assumed to be ideal; that is to say, without friction, and infinite in extent so far as the action of the steam is concerned.

The kinetic energy of a moving body is given by the formula from Mechanics: $K.\ E.=\frac{1}{2}mV^2$; the momentum by $M=mV$; and if the jet is constrained to move in a curved path, the centrifugal force $C.\ F.=\frac{mV^2}{r}$,

Where m=Mass of the body;
V=Velocity;
r=Radius of curvature of the surface.

Replacing m by its equivalent $\frac{W}{g}$, where W is the weight of the body and g is the gravitational constant, we get:

$$K.\ E.=\frac{WV^2}{2g};$$

$$M=\frac{WV}{g};$$

$$C.\ F.=\frac{WV^2}{rg}.$$

Let a jet of steam impinge perpendicularly on a plane at rest.

The momentum of the jet destroyed in the time t is $\frac{WVt}{g}$, and the impulse given to the plane in the same time is Pt, where P is the total normal pressure on the plane. Therefore, $P=\frac{WV}{g}$.

In Fig. 1, let $X-Y$ be any plane, the jet striking it perpendicularly at A. The plane moving to the right.

Let $AB=V=$Velocity of the jet in feet per second.

$AC=u=$Velocity of the plane in feet per second.

Then $CB=V_1=$Relative velocity of the jet with respect to the moving plane, or the velocity with which the jet strikes the plane, and then from above $P=\frac{WV_1}{g}=\frac{W}{g}(V-u)$.

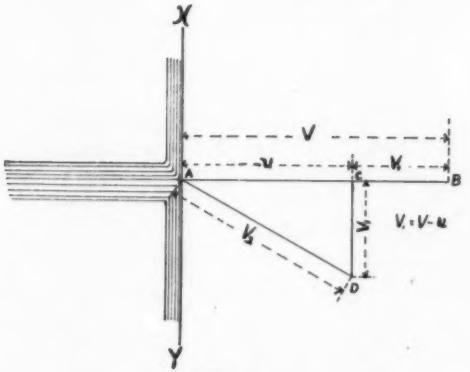


Fig. 1.

The jet is deflected so that it flows off the plane in all directions with a velocity $V-u$ relatively to the plane. Draw $CD=CB$ perpendicular to AB, then AD is the absolute velocity of the jet relative to the earth, or fixed surroundings.

Let this value be represented by V^2.

The work done per second is

$$\frac{W}{2g}(V^2-V_2^2), \text{ but } V_2^2=(V-u)^2+u^2;$$

Therefore work=

$\frac{Wu}{g}(V-u)$, and the efficiency $e=\frac{V^2-V_2^2}{V^2}=\frac{2u\,(V-u)}{V^2}$.

The velocity of the plane that gives maximum efficiency is found by differentiating the expression for efficiency, and equating the first differential coefficient to zero.

$$e=\frac{2\,u\,(V-u)}{V^2};\quad \frac{de}{du}=\frac{2V-4u}{V^2}=0.$$

$$4\,u=2V.$$

$$u=\frac{V}{2}.$$

That is, the plane should move with half the velocity of the jet and maximum efficiency=50 per cent.

In Fig. 2, let the jet enter the vane at A, the impulse is equal to $P_1=\frac{WV}{g}$, and the reaction of the jet as it leaves the vane at B is $P_1=\frac{WV}{g}$ in the direction shown. The total force P is equal to the sum of the resolved forces P_1 in the direction of P. Let θ=Total angle subtended by the vane. Then

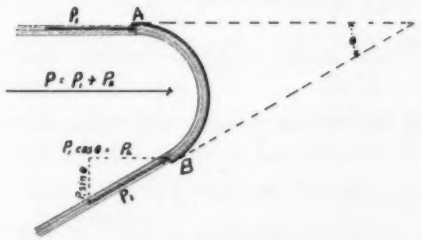


Fig. 2.

$$P=P_1+P_2=P_1+P_1\cos\theta=\frac{WV}{g}(1+\cos\theta).$$

The side thrust on the vane, at right angles to P, is

$$P_1\sin\theta=\frac{WV}{g}\sin\theta.$$

In the above example, P is taken parallel to the entering stream.

If the entering stream makes the angle α with the direction of P, and the leaving stream makes the angle β, we would have for the general case:

$$P=P_1 \cos \alpha+P_1 \cos \beta=\frac{WV}{g}(\cos \alpha+\cos \beta).$$

If the vane moves with the velocity u, then the velocity of the jet relatively to the vane is $V-u$, and substituting in the above equations, $V-u$ for V, we have:

$$P=\frac{W}{g}(V-u)(1+\cos \theta).$$

$$P=\frac{W}{g}(V-u)(\cos \alpha+\cos \beta).$$

The velocity of the vane being u, the work done per second is $Pu=\frac{Wu}{g}(V-u)(1+\cos \theta)$, which is a maximum when $u=\frac{V}{2}$; Substituting this value of u in the expression for work, we get: Work done per second $=\frac{WV^2}{4g}(1+\cos \theta)$.

Kinetic energy of the entering stream $=\frac{WV^2}{2g}$.

$$\text{Efficiency}=\frac{\text{Work done}}{K.E.}=\frac{WV^2(1+\cos \theta)}{4g}\div\frac{WV^2}{2g}=\frac{1+\cos \theta}{2},$$

and assuming the stream to enter the blade, as in Fig. 2, the efficiency is seen to depend on the exit angle θ.

$$\theta=180°,\ e=0;\quad \theta=90°,\ e=.5;\quad \theta=0,\ e=1.$$

In Fig. 3, let $AB=V=$Absolute velocity of the steam leaving the nozzle;

$BC=EF=u=$Blade velocity;

$AC=V_1=$Relative entering velocity of the steam;

$DE=V_2=$Relative exit velocity of the steam;

$DF=V_3=$Absolute exit velocity of the steam.

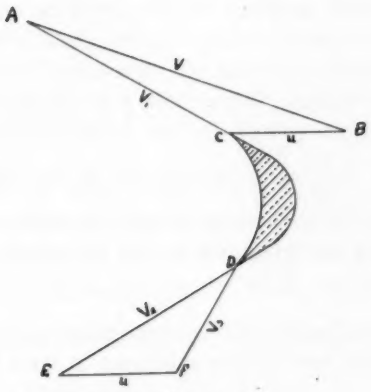


Fig. 3.

Then whatever the angles of entrance and exit may be, the energy extracted per pound of entering steam is:

$\frac{V^2}{2g}-\frac{V_1^2}{2g}+\frac{V_2^2}{2g}-\frac{V_3^2}{2g}$ and the efficiency $=\frac{V^2-V_1^2+V_2^2-V_3^2}{V^2}$.

FLOW OF STEAM THROUGH NOZZLES.

The general equation for the adiabatic flow of elastic fluids is $E_1+p_1v_1+\frac{V_1^2}{2g}=E_2+p_2v_2+\frac{V_2^2}{2g}$. The total energy of the fluid remains constant.

The subscript $_1$ indicates initial conditions and the subscript $_2$ final conditions.

E = Intrinsic energy of the fluid;
p = Pressure;
v = Specific volume at pressure p;
V = Velocity.

Then by neglecting the initial velocity V_1, which is usually very small, and rearranging, we have:

$$\frac{V_2^2}{2g}=E_1-E_2+p_1v_1-p_2v_2. \quad . \quad . \quad . \quad . \quad (1)$$

The right-hand member of this equation represents the total change in internal energy and external work done upon and by the substance during its expansion from p_1v_1 to p_2v_2, and since the changes have been due solely to the work done by the heat energy in the substance, the resulting kinetic energy, $\frac{WV_2^2}{2g}=\frac{V_2^2}{2g}$ per pound of the issuing jet, is numerically equal to the amount of heat energy each pound of the substance has given up during its expansion from p_1v_1 to p_2v_2.

Let H_1=Total heat of the substance at state point p_1v_1;
H_2=Total heat of the substance at state point p_2v_2.

Then for each pound of substance, the kinetic energy of the jet flowing from the nozzle is

$$\frac{V_2^2}{2g}=(H_1-H_2)\times 777.5 \text{ foot pounds};$$

$$V_2^2=2g\times 777.5\times(H_1-H_2);$$

$$V_2=\sqrt{2g\times 777.5\times(H_1-H_2)};$$

$$=223.7\sqrt{H_1-H_2}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

From this the theoretical exit velocity of the jet can be computed.

1 B.T.U.=777.5 foot-pounds of energy. $g=32.17$.

If a gas expands adiabatically; that is, without the addition or extraction of heat as heat, the equation of the curve of expansion is $pv^k=p_1v_1^k=p_2v_2^k$, etc., where k is the ratio of the specific heat of the gas at constant pressure to that at constant volume$=\frac{c_p}{c_v}$.

Let a quantity of gas be at the state point p_1v_1, and expand adiabatically to p_2v_2. The external work done is

$$W=\int_{v_1}^{v_2} pdv = p_1 v_1^{k} \int_{v_1}^{v_2} \frac{dv}{v^k} =$$

$$\left(\frac{1}{v_2^{k-1}}-\frac{1}{v_1^{k-1}}\right)\frac{p_1 v_1^{k}}{k-1}=\frac{p_1 v_1}{k-1}\left\{1-\left(\frac{v_1}{v_2}\right)^{k-1}\right\} \quad . \quad (3)$$

No heat being added, the external work is limited in amount to the intrinsic energy at $p_1 v_1$. Expanding to infinitely low pressure p_2, v_2 becomes infinitely large, so that $\frac{v_1}{v_2}=0$, and the expression for the work becomes, $W=\frac{p_1 v_1}{k-1}$, which is the total intrinsic energy of the gas at $p_1 v_1$, and is equal to E_1. Consequently, $E_2=\frac{p_2 v_2}{k-1}$.

Equation (1) can now be written

$$\frac{V_2^2}{2g}=\frac{p_1 v_1}{k-1}-\frac{p_2 v_2}{k-1}+p_1 v_1-p_2 v_2=\frac{k}{k-1}(p_1 v_1-p_2 v_2). \quad . \quad (4)$$

Since $p_1 v_1^{k}=p_2 v_2^{k}$; $p_2 v_2=p_1 v_1\left(\frac{v_1}{v_2}\right)^{k-1}=p_1 v_1\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}}$,

Therefore, $\frac{V_2^2}{2g}=p_1 v_1\left(\frac{k}{k-1}\right)\left\{1-\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}}\right\}$

$$V_2=\sqrt{2g p_1 v_1\left(\frac{k}{k-1}\right)\left\{1-\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}}\right\}}. \quad . \quad . \quad (5)$$

Let A_0=Area of orifice in square feet.

Then the volume flowing per second is $A_0 V$, and the weight F, flowing per second is $\frac{A_0 V}{v_2}$.

But $v_2=v_1\left(\frac{p_1}{p_2}\right)^{\frac{1}{k}}$; Therefore, the equation for flow is:

$$F=A_0\sqrt{\frac{2gk}{k-1}\left(\frac{p_1}{v_1}\right)\left\{\left(\frac{p_2}{p_1}\right)^{\frac{2}{k}}-\left(\frac{p_2}{p_1}\right)^{\frac{k+1}{k}}\right\}}. \quad . \quad . \quad (6)$$

Which is Zeuner's equation for the flow of gases.

F=Flow of gas in pounds per second.

A_0=Area of orifice in square feet.

k=Ratio of specific heats=$\frac{c_p}{c_v}$.

p_1=Initial pressure in pounds per square foot.

p_2=Final pressure in pounds per square foot.

v_1=Initial specific volume in cubic feet per pound.

v_2=Final specific volume in cubic feet per pound.

Equation (6) is evidently a maximum when the part in brackets $\left\{ \left(\frac{p_2}{p_1}\right)^{\frac{2}{k}} - \left(\frac{p_2}{p_1}\right)^{\frac{k+1}{k}} \right\}$ is a maximum.

Differentiating this expression with respect to p_2 (p_1 being regarded as constant), and equating the first differential coefficient to zero, we get:

$$\frac{p_2}{p_1} = \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

p_2 is now the pressure at the throat (smallest cross section) of the nozzle. Replacing it by p_0 and substituting in the above equation of Zeuner's (6), we get:

$$F = A_0 \sqrt{\frac{2gk}{k-1} \left(\frac{p_1}{v_1}\right) \left(\frac{p_0}{p_1}\right)^{\frac{2}{k}}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

The different letters having the same significance as before.

For initially dry saturated steam, k has the value 1.135, and for highly superheated steam this increases to 1.34 or more.

The following table gives the flow of steam in pounds per second for the different values of k. The value of $\frac{p_0}{p_1}$ being obtained from equation (7), by substituting the different values of k. When using these equations for the flow of steam, the back pressure must be as low or lower than p_0.

k	$\frac{p_0}{p}$	F
1.135	.5774	$3.604\ A_0\sqrt{\frac{p_1}{v_1}}$
1.20	.5645	3.678 " "
1.25	.5549	3.732 " "
1.30	.5457	3.784 " "
1.333	.5398	3.817 " "

Expressing A_0 in square inches and p in pounds per square inch, the equations for flow become:

k	F
1.135	$.3003\ A_0\sqrt{\frac{p_1}{v_1}}$
1.20	.3065 " "
1.25	.3110 " "
1.30	.3153 " "
1.333	.3181 " "

The value of $\frac{p_0}{p_1}$ remains constant regardless of what the initial or back pressure may be, provided the back pressure is as low or lower than p_0. For initially dry saturated steam the equation for the flow is $F=.3003\ A_0\sqrt{\frac{p_1}{v_1}}$.

Grashof gives the equation $F=.01654\ A_0\ p_1^{.9696}$.

Napier gives the equation $F=\frac{A_0 p_1}{70}$.

Grashof's equation is very nearly expressed by $F=\frac{A_0 p_1^{.97}}{60}$, which practically agrees with Zeuner's equation for initially dry saturated steam. This last equation may be written $A_0=\frac{60F}{p_1^{.97}}$, which gives the area in square inches necessary to pass a given amount of steam per second. Zeuner's theoret-

ical equation may be expressed by $F=K\sqrt{\frac{p_1}{v_1}}$ for initially dry saturated or superheated steam (provided you can obtain the initial specific volume). For wet steam, neglecting the volume of water present, which is small, this equation becomes $F=K\sqrt{\frac{p_1}{x_1 v_1}}$, which shows that the flow of wet steam is inversely proportional to the square root of the initial quality x_1.

Grashof's formulae may now be written

$$F=\frac{A_0 p_1^{.97}}{60\sqrt{x_1}};\quad A_0=\frac{60F\sqrt{x_1}}{p_1^{.97}};$$

for $x_1=1$, these equations become the same as those given before.

For Grashof's formulae, a curve can be drawn showing the values of $\frac{60}{p_1^{.97}}$, which will simplify the computations. None of these equations are of any use, however, unless the pressure at the throat drops to about .58 of the initial pressure or lower, which in the Curtis turbine is only in the first stage and a few of the last stages. For the Curtis turbine, the following method is used: Locate the point of initial steam condition on a total heat-entropy chart, also locate the point of steam condition in the throat. The latter point is on the pressure line corresponding to the throat pressure, and at the same entropy as the initial point; the expansion in the nozzle being assumed adiabatic (isentropic).

Then take off the "heat drop" in the throat, equal to the difference in total heat between the two points, and calculate the throat velocity from the formula $V=223.7\sqrt{H_1-H_2}$, and with a velocity V calculate the volume flowing per second $=\frac{A_0 V}{144}$,

Where A_0=Area of the throat in square inches;
V=Velocity in feet per second;
v_2=Actual specific volume of the steam in the throat.

Then the flow in pounds per second$=\frac{A_0 V}{144 v_2}$.

For initially superheated steam we must have steam tables such as those of Marks and Davis, which give the properties of superheated steam. If these are not available, the specific volume may be worked out by means of the following equation, modified from that of Knoblauch, Linde and Klebe:

$$pv=0.5962T-p(1+0.0014p)\left(\frac{150,300,000}{T^3}-0.0833\right),$$

Where p=Initial pressure in pounds per square inch;
T=Initial absolute temperature;
v=Initial specific volume.

It is much simpler to modify Grashof's formulae as follows:

$$F=\frac{A_0 p_1^{.97}}{60(1+.00065D)};\quad A_0=\frac{60F(1+.00065D)}{p_1^{.97}};$$

Where D is the superheat in degrees Fahrenheit.

BLADES.

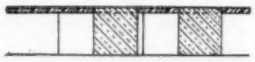
Fig. 4 shows the method of laying out the blades of a Curtis turbine.

The pitch p is decided upon (between $\frac{1}{2}$ inch and 1 inch), about $\frac{3}{4}$ inch being good practice. The width w is also decided, p being from .6 to .7 of w. The pitch and the axial width of the blades usually increase when the length of the blades is increased.

Lay down points 1 and 2, as the edges of two successive blades, distance apart p, and through them draw lines making the angle α (=entrance angle of the blades) with the center line of the blades.

Through point 2 draw a line perpendicular to the line through point 1; where this perpendicular intersects the cen-

ter line (5) is the center for describing the circular back of the first blade and the circular face of the second blade. Locate points 3 and 4 perpendicularly below 1 and 2 and distant w from them. Through 3 and 4 draw lines making the angle $\beta = \alpha$ (=exit angle of the blades) with the center line. Complete the construction by drawing in the arcs. In laying out the blades the angles α and β are made equal. If they are not to be equal on the blades, draw the angles α and β equal to $\frac{1}{2}(\alpha+\beta)$, and when the ends are milled to fit the channel base the milling cutter should be set at an angle equal to $\frac{1}{2}(\alpha-\beta)$ from the center, so that when the blades are mounted they will be at the proper angles.



SECTION ON CENTER LINE.

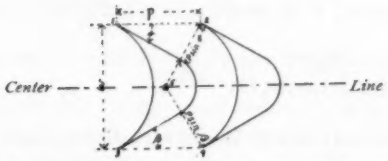


Fig. 4.—SHOWING CONSTRUCTION OF BLADES.

p = Pitch of blades.
w = Width of blades.
α = Entrance angle of blades.
β = Exit angle of blades.
$p \sin \alpha$ = Perpendicular width of steam passage at entrance.
$p \sin \beta$ = Perpendicular width of steam passage at exit.

This method of construction gives practically a constant perpendicular width for steam passage ($p \sin \alpha$ or $p \sin \beta$) between the blades. This method should not be used if the angles α and β differ by a large amount. But as in practice they differ by only a few degrees, it is permissible to use it. Theoretically the width of the steam passage should be least at the middle of the blade, owing to the centrifugal action of the steam.

The clear area through the blades at entrance is $ph \sin \alpha$, and at exit is $ph \sin \beta$, where p is the pitch of the blades and h the radial height of the blades. The relative velocity of the steam is less at exit than at entrance, owing to friction of the steam on the blades, so that for equal angles of entrance and exit the height at exit should be greater than that at entrance, and as the exit angle is usually made smaller than the entrance angle this height is increased still more. The end of the blade should be inclined, that is, the exit side should be longer than the entrance side. The increase in length being equally divided on each side of the center line of the blades.

NOZZLES.

As shown before, the pressure at the throat of a nozzle is about .58 of the initial pressure for initially dry saturated steam, and as it is desirable not to have too great a pressure in the casing, nor a large temperature range from one end of the turbine to the other, the nozzles for the first stage are so made as to expand the steam to a lower pressure than .58 of the initial pressure. This is done by having the nozzles themselves expand; that is, increase in cross-sectional area from the throat to the mouth, the amount of increase depending on the final pressure desired in the first stage. The length of the nozzle necessary also depends on the amount of expansion desired, the expansion of the linear dimensions of the nozzle being about 1 in 12.

The nozzles for the intermediate stages are made non-expanding; that is, the walls of the nozzles are parallel, the pressure in the throat being the same as the back pressure.

It is better that the first-stage nozzles under-expand than over-expand, as it has been determined from experiments that the loss from under-expansion is less than from the same amount of over-expansion.

Fig. 5 is a sketch of an expansion nozzle. For Curtis turbines, the cross section at the throat t is a square or rectangle with rounded corners.

The radial height of the nozzle is kept practically constant, so that all expansion is lateral. A nozzle could not be used that was cut off at the end of the expanding portion (a—c in sketch) at right angles to the axis, it being obvious that an extension or "tail" a—b is necessary to properly direct the steam on the blades.

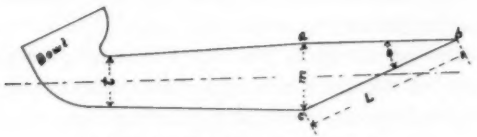


Fig. 5.—EXPANSION NOZZLE.

t = Throat.
bc = Mouth.
m = End of expansion.

$$\text{Expansion ratio} = \frac{\text{Area of } m}{\text{Area of } t}$$

In order to avoid spreading the jet and making the ratio of expansion uncertain, this tail is often made non-expanding; that is, with walls parallel to the axis of the nozzle, as shown in Fig. 5.

From Fig. 5, $L = m \operatorname{cosec} \theta$; where θ = angle between the axis of the nozzle and the plane of the wheel.

m = Width of the nozzle at the end of expanding portion.

L = Width of the nozzle at the mouth.

Knowing the amount of steam passing through the turbine in a given time, we can calculate the total throat area necessary, and then decide upon the number of nozzles.

Let L = Length of the mouth b—c in inches; (Fig. 5.)
n = Number of nozzles;
h = Height of blades in inches;
a = Entrance or exit angle of blades;
V_1 = Relative velocity of the steam in the blades.

Then $\dfrac{n \times L \times h \times \sin a}{144}$ = Net area between the blades for

the passage of steam in the arc occupied by the nozzles. This expression multiplied by the relative velocity of the steam, gives the volume of steam passing per second.

If F=Weight of steam flowing through the turbine per second, and

v=Specific volume of the steam ($=vx$ for wet steam), then we have

$$Fv = \text{Volume of steam flowing per second};$$

$$= \frac{n \times L \times h \times V_1 \times \sin\alpha}{144};$$

or

$$h = \frac{F \times v \times 144}{n \times L \times V_1 \times \sin\alpha};$$

which gives the neeessary height of the blades. As noted before, this height should be calculated for both the entrance and exit sides of the blades, ample allowance being made to prevent the steam spilling over the ends of the blades.

The blades for the first row should be about 1.5 inches in height, which limits the radial height of the nozzle. This holds good only to about the fifth stage, where the nozzles completely fill the circumference of the diaphragm; beyond the fifth stage, the height of the nozzle is increased, and therefore the length of the blades must be increased.

When the nozzles are increased in size, it is a good policy not to increase the width of the nozzle any more than necessary, but to increase the radial height, keeping the nozzle comparatively narrow.

The length of the mouth L should be kept small, as this reduces the loss due to eddy currents, which are set up in the jet of steam as soon as the jet leaves one of the restraining walls of the nozzle.

Having gone over the theory of the impulse turbine and the flow of steam through nozzles, I will now work out the design for the principal parts of a turbine with the following assumed specifications:

Specifications.

Number of stages ahead,	9
Pitch diameter, inches,	144
Revolutions per minute,	230
Shaft horsepower,	14,500
Initial steam conditions,	180 lbs. abs. and 100° F. superheat.
Final steam conditions,	1.5 pounds absolute.
Rows of buckets on 1st stage,	4
Rows of buckets on 2d to 9th stages,	3

Energy.

B.T.U. at 180 pounds and 100° superheat,	1,254.3
B.T.U. after adiabatic expansion to 1.5 lbs. abs.,	924.0
Total available energy in B.T.U. per pound,	330.3

Bucket speed $=\frac{230\times 12\times\pi}{60}=145$ feet per second (about).

Desired pressure in shell of 1st stage, lbs. abs.,	70
B.T.U. in steam after adiabatic expansion from initial conditions to 70 pounds, absolute,	1,171.3
B.T.U. drop in 1st stage nozzles,	83.0
Theoretical jet velocity for 1st stage, foot-seconds, $=227.3\sqrt{83}=$	2,038
Loss in velocity from curve, foot-seconds,	138
Actual exit velocity, foot-seconds,	1,900
Nozzle angle, degrees,	20

From the velocity diagram for the first stage, we get the efficiency to be 47.3 per cent., and the B.T.U. used $=83\times 47.3$ per cent. $=39.3$. The remainder of the B.T.U. being given back into the steam as reheat.

B.T.U. in 1st stage receiver $=1,254.3-39.3=1,215.0$.

Steam condition in the 1st-stage receiver 70 pounds absolute and 68 degrees superheat (pressure 70 pounds and total heat 1,215.0 B.T.U.).

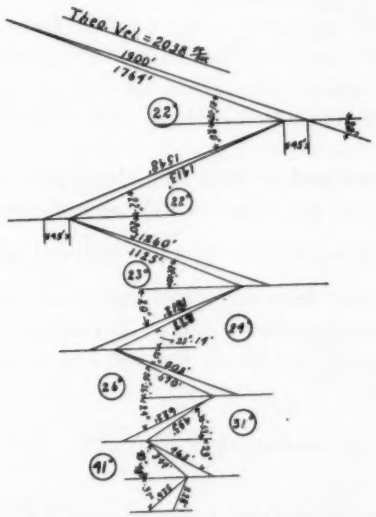


Fig. 6.—VELOCITY DIAGRAM, FIRST STAGE.

NOTE.—Figures inclosed in circles are the entrance angles of the blades.

$$\text{Eff.} = \frac{1{,}965{,}098}{2{,}038^2} = 47.3 \text{ per cent.}$$

Extractions: $1{,}900^2 - 1{,}764^2 = 498{,}304$
$1{,}548^2 - 1{,}413^2 = 399{,}735$
$1{,}260^2 - 1{,}125^2 = 321{,}975$
$1{,}012^2 - 877^2 = 255{,}015$
$803^2 - 670^2 = 195{,}909$
$622^2 - 493^2 = 143{,}835$
$463^2 - 344^2 = 96{,}033$
$326^2 - 228^2 = 54{,}292$

$$\frac{1{,}965{,}098}{2g} = 30{,}544 \text{ ft. lbs.}$$

B.T.U. in steam after adiabatic expansion from 70 pounds abs. and 68 degrees superheat to 1.5 pounds abs., 956.0
B.T.U. available for 2—9 stages = 1,215.0 — 956.0 = 259.0

Equally divided between eight stages gives 32.4 B. T. U. per stage. Owing to the fact that the constant-pressure lines

on the total heat-entropy chart are not parallel, but diverge, we would not arrive at the final pressure desired, 1.5 pounds absolute, if we used a heat drop of 32.4 B.T.U. per stage, but would be at a higher pressure.

It is therefore necessary to use a reheat factor by which the theoretical heat drop must be multiplied to give the necessary heat drop per stage. This could be derived by a cut-and-try process on the total heat-entropy chart, but it is long and tedious.

* "It will be found by trial and adjustment that if the theoretical heat drop per stage, which we may designate by the symbol $\frac{\triangle H}{n}$, be multiplied by the empirical factor $1+K$, we will have the heat drop per stage which will give an actual equality in the quantity of energy developed in each stage. In the above empirical factor the value of K is found by the equation

$$K=0.00056\left(\frac{n-1}{n}\right)\triangle H(1-E).$$"

$\triangle H$=Total available heat drop=259.0 B.T.U.;
n=Number of stages=8;
E=Probable internal efficiency of each stage=about 61 per cent.;
$K=0.00056\times\frac{7}{8}\times 259\times 0.39=0.04949$;
Reheat factor=$1+K=1.04949$.
Heat drop per stage=$32.4\times 1.04949=34.0$ B.T.U.

The method of dividing the energy equally between the different stages is much simpler than to separately design each stage, allowing the pressure to drop to about .58 of the pressure in the receiver of the preceding stage. In the former case the pressure in the throat is equal to the back pressure in the stage, and for the first half of the stages is usually higher than .58 of the pressure in the preceding stage, and for

*Forrest E. Cardullo, in the "Journal of the American Society of Mechanical Engineers," February, 1911.

the last half is equal to or lower than .58 of the preceding pressure.

Theoretical jet velocity for 2–9 stages, foot-seconds,

$$= 223.71\sqrt{34} = 1{,}305$$

Loss in velocity from curve, foot-seconds, . . . 61

Actual nozzle velocity, foot-seconds, . . . 1,244

Nozzle angles are all 20 degrees, except for the last stage, where they are increased to 25 degrees to allow freer passage of the wet steam.

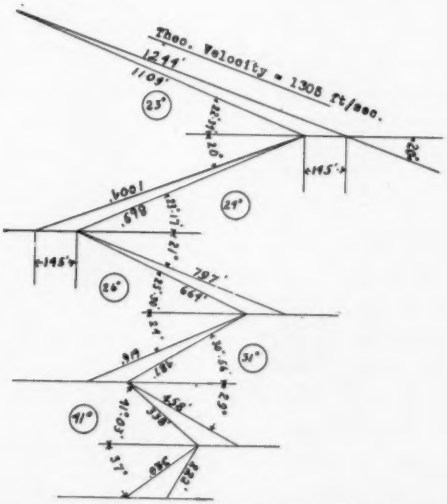


Fig. 7.—Velocity Diagram for Second to Eighth Stages, Inclusive.

Note.—Figures inclosed in circles are the entrance angles of the blades.

$$\text{Eff.} = \frac{1{,}055{,}746}{1{,}305^2} = 62 \text{ per cent.}$$

Extractions: $-\ 1{,}244^2 - 1{,}109^2 = 317{,}655$
$1{,}004^2 - 869^2 = 252{,}855$
$797^2 - 664^2 = 194{,}313$
$616^2 - 487^2 = 142{,}287$
$458^2 - 338^2 = 95{,}520$
$320^2 - 222^2 = 53{,}116$

$$\frac{1{,}055{,}746}{2g} = 16{,}410 \text{ ft. lbs.}$$

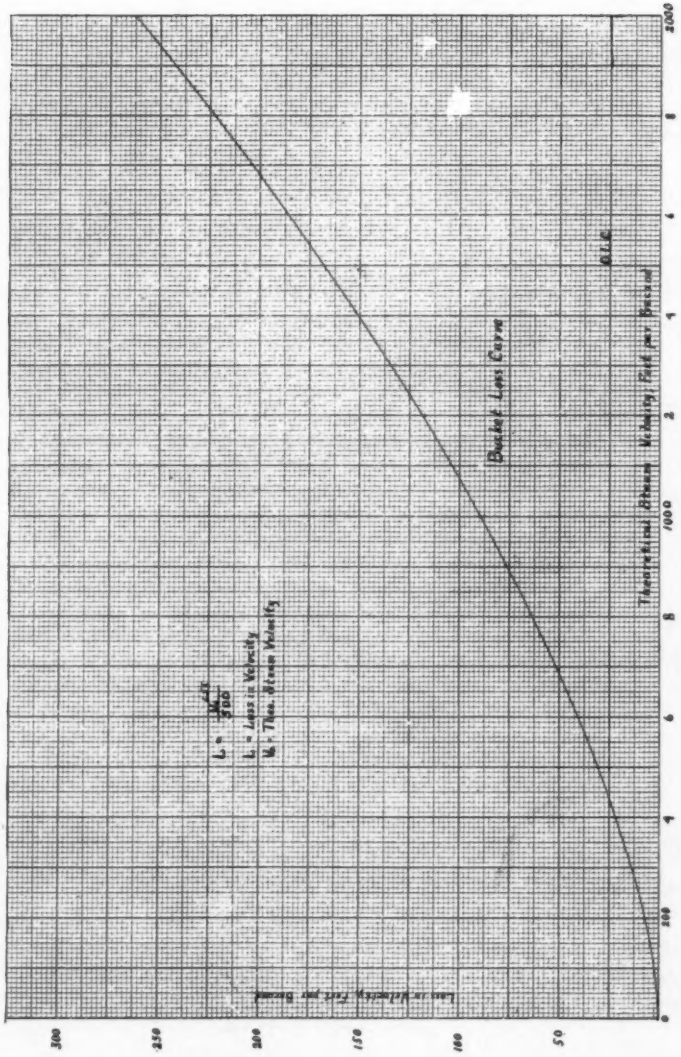


Fig. 9.—Bucket Loss Curve.

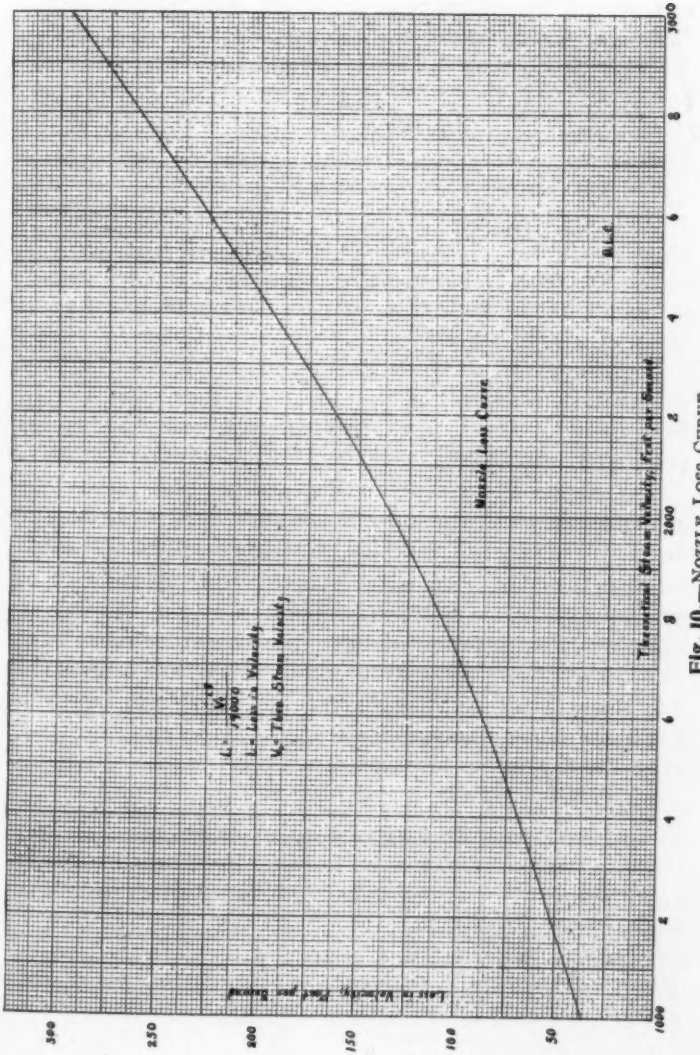


Fig. 10.—Nozzle Loss Curve.

From the velocity diagram for the 2d to 8th stages, inclusive, we get the efficiency to be 62 per cent., and the useful B.T.U. to be 34×62 per cent. $= 21.1$.

The curves giving the loss in velocity are derived from experimental data, the loss being proportional to the nth power of the velocity, so that the equation of the curve is of the form $\text{Loss} = kV^n$.

The curves given are not for any particuiar turbine, but are only approximate. These curves should be obtained from an actual turbine of similar construction. The blade loss is shown the same for fixed as for moving blades, but experiments should be made to determine whether or not the loss is not greater in moving blades than in fixed blades.

High superheat decreases the loss due to friction as well as the rotation losses. It is, therefore, economical to use superheat, though more than 150 degrees superheat does not appear to be beneficial.

The higher the pressure and superheat, the greater the available energy per pound of steam, but it will only be from an extensive series of experiments that we will be able to determine the most economical pressure and superheat to be used on any particular type of turbine.

In designing a turbine of this type it is necessary to have an accurate total heat-entropy chart, made to a large scale. The steam tables can be used, but they require an enormous amount of labor, whereas with the chart it is a simple matter, as all the expansions are isentropic.

Nozzle Calculations for the First Stage.

Initial steam, . 180 pounds absolute, 100 degrees superheat.	
Throat pressure, pounds, . .	$180 \times .5555 = 100$
First-stage receiver pressure, pounds, ...	70
B.T.U. per pound of initial steam, . . .	1,254.3
B.T.U. per pound after adiabatic expansion to 100 pounds absolute, ...	1,200.8

B.T.U. per pound after adiabatic expansion to 70 pounds absolute, ... 1,171.3
B.T.U. given up in throat, ... 53.5
nozzle, ... 83.0
Velocity of steam in throat, ft.-secs., $223.7\sqrt{53.5}$ = 1,636
at mouth, ft.-secs., $223.7\sqrt{83.0}$ = 2,038
Quality of steam in throat, superheat, degrees, . 26
at mouth, ...991
Specific volume of steam in throat (100 pounds, and 26 degrees superheat), ... 4.62
Specific volume of steam at mouth (70 pounds, quality .991), ... 6.144

To find the expansion ratio the following formula is used:

$$\frac{A_m}{A_t}=\frac{V_t}{V_m}\times\frac{v_m}{v_t}.$$

A_m = Area of mouth;
A_t = Area of throat.
V_m = Velocity at mouth.
V_t = Velocity in throat.
v_m = Actual specific volume at mouth.
v_t = Actual specific volume in throat.

$\frac{A_m}{A_t}=\frac{1,636\times 6.144}{2,038\times 4.62}=1.07$ = Expansion ratio of first-stage nozzles.

For a turbine of the size and design above we are safe in assuming an overall full-load water rate of about 14 pounds, which gives a flow of 56.4 pounds per second through the first-stage nozzles, and allowing a loss of 5 per cent. through the diaphragm bushings, gives a flow of 53.6 pounds per second through the other nozzles.

Total area required for throat of first-stage nozzles,

square inches = . . . $\frac{56.4\times 4.62\times 144}{1,636}=22.93$

Total area at end of expansion, sq. ins. $=22.93\times 1.07=24.53$
Use 16 nozzles, area at throat of each, square inches = 1.433

Area at end expansion of each, square inches,= . 1.533
Nozzles to be 1.3 inches high by 1.1 inches wide.
Width of nozzles at end of expansion, inches,= 1.18
Length of nozzle mouth, inches=1.18 cosec 20°= 3.45

Allowing a thickness of 0.1 inch for the walls at the mouth of the nozzle and four extra nozzles for overload, gives a length of nozzle opening of about 71 inches=about 55 degrees arc for first-stage nozzles.

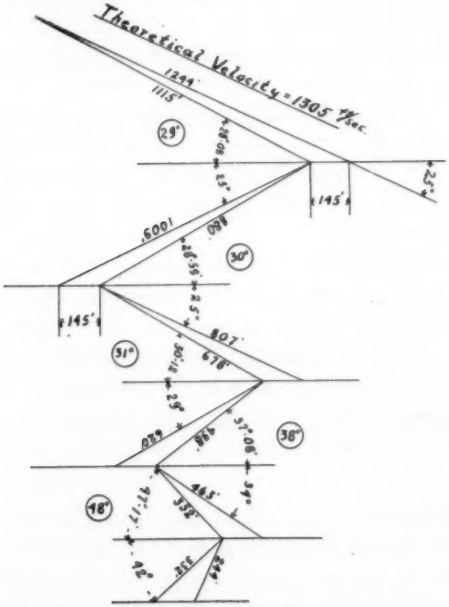


Fig. 8.—VELOCITY DIAGRAM FOR NINTH STAGE.

Figures inclosed in circles are the entrance angles of the blades.

Extractions: $-1{,}244^2 - 1{,}115^2 = 304{,}311$
$1{,}009^2 - 880^2 = 243{,}681$
$807^2 - 678^2 = 191{,}565$
$620^2 - 498^2 = 136{,}396$
$463^2 - 352^2 = 90{,}465$
$332^2 - 244^2 = 50{,}688$

$$\text{Eff.} = \frac{1{,}017{,}106}{1{,}305^2} = 59.7 \text{ per cent.}$$

$$\frac{1{,}017{,}106}{2g} = 15{,}809 \text{ ft. lbs.}$$

For the second and succeeding stages, the same general method is used, increasing the height of the nozzle (within practical limits) in preference to increasing the width, when it becomes necessary to increase the size of the nozzles (sixth stage).

From the second to fifth stages, inclusive, use nozzles 1.8 inches high by 0.9 inch wide. For the remaining stages, the height must be increased as the nozzles completely fill the circumference of the diaphragm.

The exit height of the blades is calculated by the formula given before :

$$h=\frac{144\times F\times v}{n\times L\times V_1\times \sin\alpha}.$$

Exit height of 1st row of blades, 1st stage, ins.,	1.72 (moving).
2d	2.15 (fixed).
3d	2.72 (moving).
4th	3.30 (fixed).
5th	3.80 (moving).
6th	4.35 (fixed).
7th	5.10 (moving).

From the velocity diagrams it is seen that the first few rows do practically all of the work, so in order to reduce the height of the last row of blades, the angles of these blades should be increased over that shown on the diagram. This would reduce the theoretical efficiency somewhat, but at the same time it reduces the windage loss due to long blades, and gives a freer passage for the slow-moving steam through these blades.

TOTAL HEAT–ENTROPY CHART.

In the accompanying total heat-entropy chart the horizontal lines are lines of constant total heat ; the vertical lines are lines of constant entropy ; the curves slanting up and to the right are lines of constant pressure ; the curves slanting down and to the right are lines of constant quality or constant superheat.

The saturation curve is marked on the diagram and gives the steam properties of saturated steam at the various pressures. The part of the diagram below the saturation curve gives the properties of wet steam, and the part above the saturation curve gives the properties of superheated steam.

In designing an impulse turbine, the point representing the initial steam condition (A=180 pounds and 100 degrees superheat) is first plotted, and then the point representing the exhaust conditions, assuming adiabatic expansion (P=1.5 pounds). The latter point is on the pressure line corresponding to the exhaust pressure and at the same entropy as the initial point.

The distance between A and P is the total available energy per pound of steam in B.T.U.=1,254.3—924.0=330.3.

The pressure desired in the first stage being 70 pounds, locate point B at the intersection of the 70-pounds pressure line and the entropy line passing through A. The distance from A to B gives the B.T.U.'s available for the first stage =1,254.3—1,171.3=83.0. From the velocity diagram for the first stage, we found that only 39.3 B.T.U. were changed into useful work, the remainder being given back into the steam as re-heat. The total heat remaining in the steam after it has passed through the first stage=1,254.3—39.3=1,215.0 B.T.U. The intersection of the 70-pound line and the 1,215.0 B.T.U. line locates B^1, the steam condition in the first stage receiver.

B^1P^1 is the available energy per pound of steam left in the steam after it has passed through the first stage and equals 1,215.0—956.0=259.0 B.T.U. to be equally divided between eight stages. By means of the reheat factor we found that there should be 34.0 B.T.U. drop per stage, and from the velocity diagram for the second—eighth stages, we get the efficiency to be 62 per cent. or a reheat of 38 per cent.=12.9 B.T.U.

Similarly for the ninth stage we get an efficiency of 59.7 per cent., and a reheat of 13.7 B.T.U.

The various points are located on the total heat-entropy chart, as follows:

Stage.	B.T.U.	Bowl pressure.	Bowl conditions.	Throat pressure.	Throat conditions.	B.T.U. in throat.	Throat velocity.	Sp. Vol. in throat, dry.	Actual Sp. Vol. in throat.	Steam per sq. in. of throat.	Total nozzle area	Net nozzle area.	Sp. Vol. in receiver.	Expansion ratio.
	1,254.3 83.0	*pounds.*	*degrees.*	*pounds.*	*degrees.*									
1	1,171.3 43.7	180	100	100	26	53.5	1,636	4.43	4.62	2.46	22.93	22.93	6.88	1.07
	1,215.0 34.0													
2	1,181.0 12.9	70	68	47	18	34.0	1,305	9.02	9.28	.977	57.75	54.86	9.64	1.00
	1,193.9 34.0													
3	1,159 9 12.9	47	43	30.8	.996	34.0	1,305	13.41	13.36	.679	83.10	78.94	13.78	1.00
	1,172.8 34.0													
4	1,138.8 12.9	30.8	17	19.75	.982	34.0	1,305	20.33	19.96	.454	124.20	118.0	20.25	1.00
	1,151.7 34.0													
5	1,117.7 12.9	19.75	.996	12.3	.970	34.0	1,305	31.62	30.67	.296	190.9	181.4	31.08	1.00
	1,130.6 34.0													
6	1,096.6 12.9	12.3	.983	7.6	.959	34.0	1,305	49.60	47.56	.191	296.0	281.2	48.21	1.00
	1,109.5 34.0													
7	1,075.5 12.9	7.6	.972	4.5	.948	34.0	1,305	81.00	76.78	.118	477.8	453.9	77.76	1.00
	1,088.4 34.0													
8	1,054 4 12 9	4.5	.960	2.6	.936	34.0	1,305	135.4	126.7	.072	788.5	749.1	128.5	1.00
	1,067.3 34.0													
9	1,033.3 13,7	2.6	.949	1.5	.925	34.0	1,305	227.1	210.1	.043	1,307.2	1,242.0	213.3	1.00
	1,047.0		.939											

C is the steam condition in the throat of the second stage nozzles, and is at the same entropy as B^1, but has a total heat of 34.0 B.T.U. less than B^1.

C^1 is on the same pressure line as C, but has a total heat 12.9 B.T.U. greater than C; C^1 represents the steam condition in the second-stage receiver.

The other points are similarly located, finally arriving at the exhaust condition at J^1.

THEORETICAL WATER RATE.

For every pound of steam flowing through the turbine, the first stage extracts 30,544 foot-pounds of energy. The second to eighth stages, inclusive, extract 16,410 foot-pounds each, less 5 per cent. for assumed bushing loss. This gives a total for the second to eighth stages of 109,026.5 foot-pounds.

The ninth stage extracts 15,809 foot-pounds, less 5 per cent. = 15,118.5 foot-pounds.

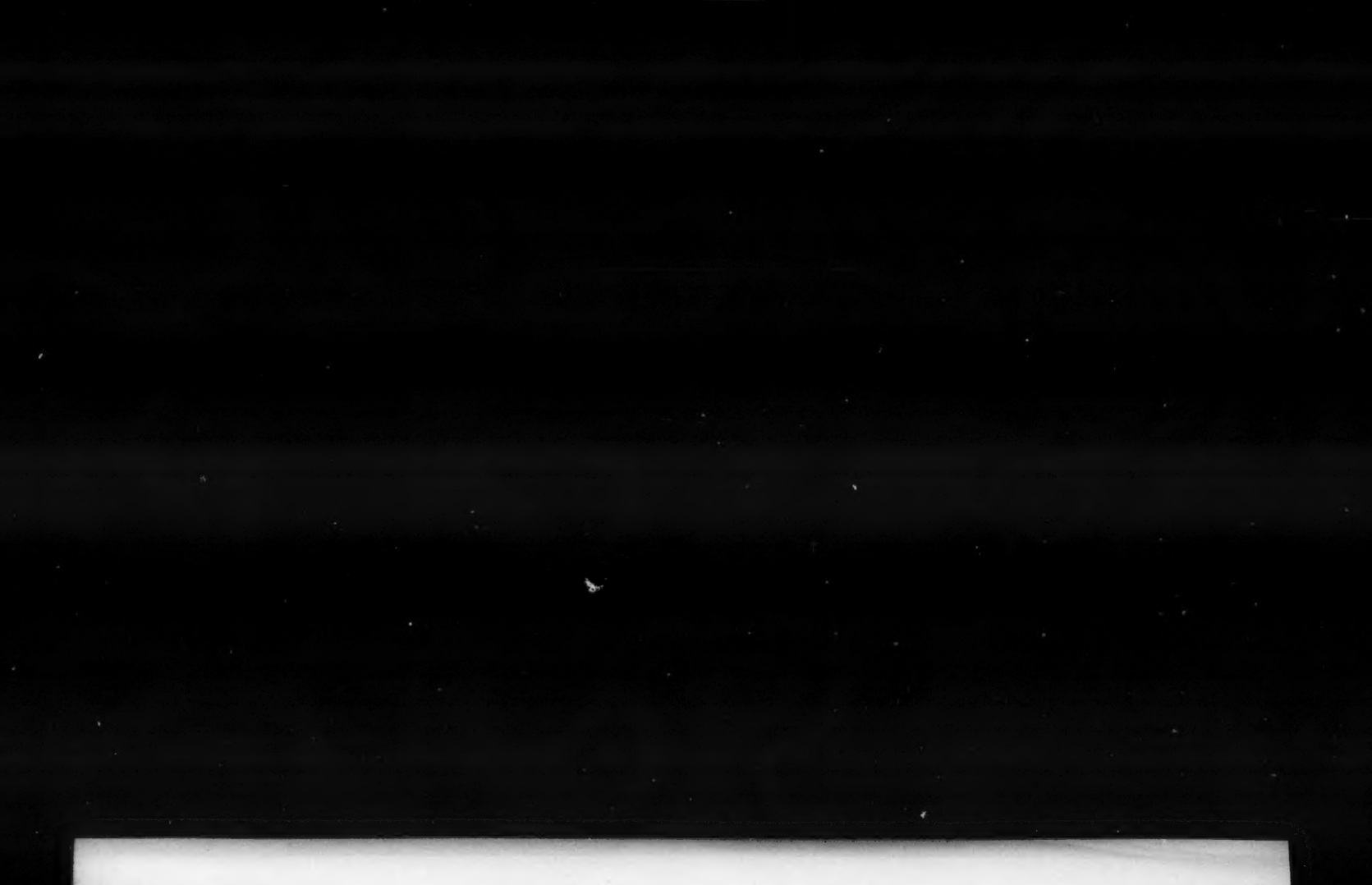
Total energy extracted per pound of steam flowing through the turbine = 30,544 + 109,026.5 + 15,118.5 = 154,689 foot-pounds.

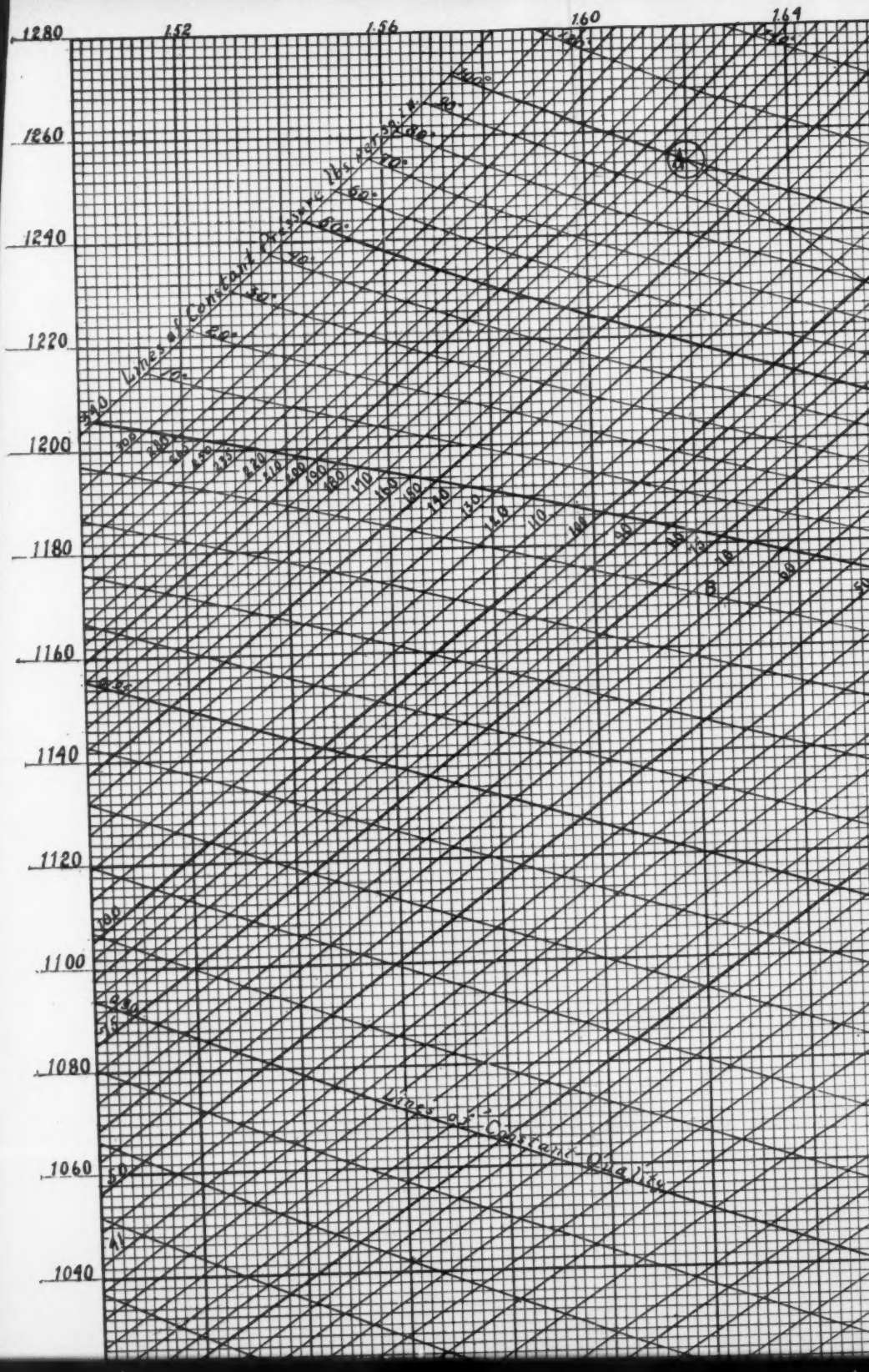
1 H.P. = 33,000 × 60 = 1,980,000 foot-pounds per hour.

$$\text{Theoretical water rate} = \frac{1{,}980{,}000}{154{,}689} = 12.8 \text{ pounds per S.H.P. per hour.}$$

Allowing 10 per cent. for frictional and other losses gives an actual water rate of 14.1 pounds per S.H.P. per hour.

NOTE.—In drawing the velocity diagrams, they are first solved trigonometrically, and then drawn to scale.




1280
1260
1240
1220
1200
1180
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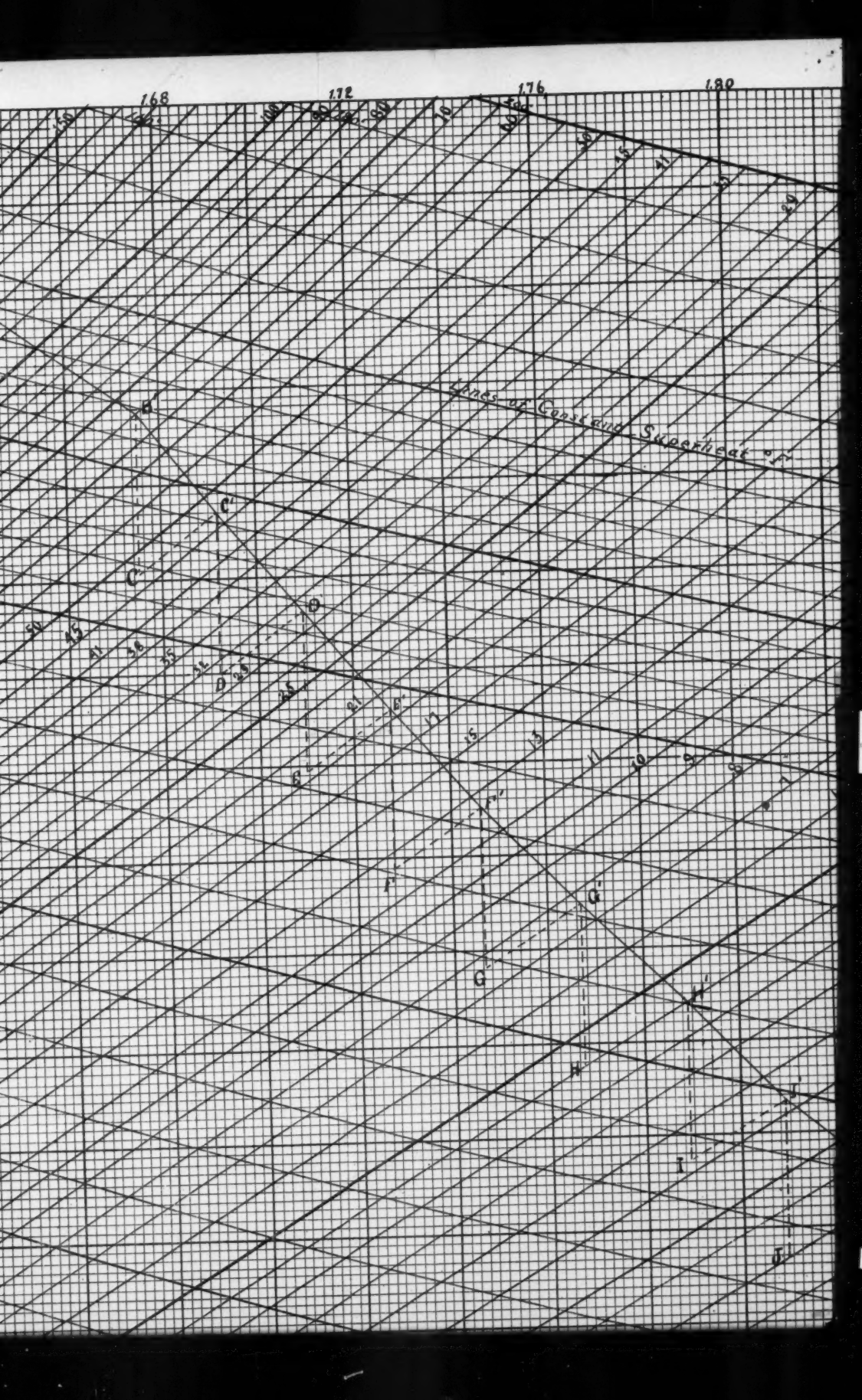
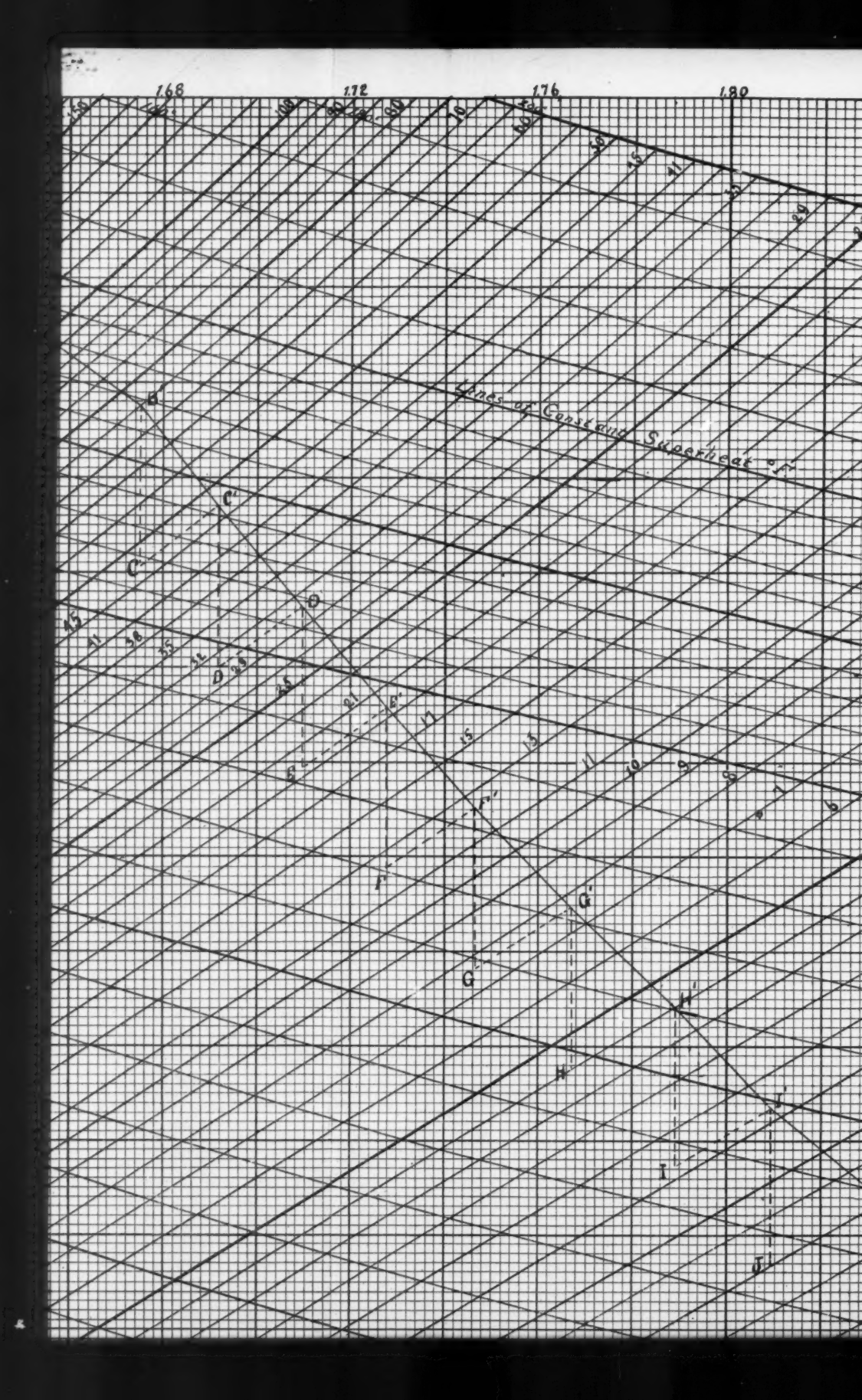
1120
1100
1080
1060
1040
152
156
160
164
Lines of Constant Pressure
Lines of Constant Quality


168
172
176
180
Lines of Constant Superheat °F
C'
C
D
E
F
G
H
I
J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Lines of Constant Superheat °F.
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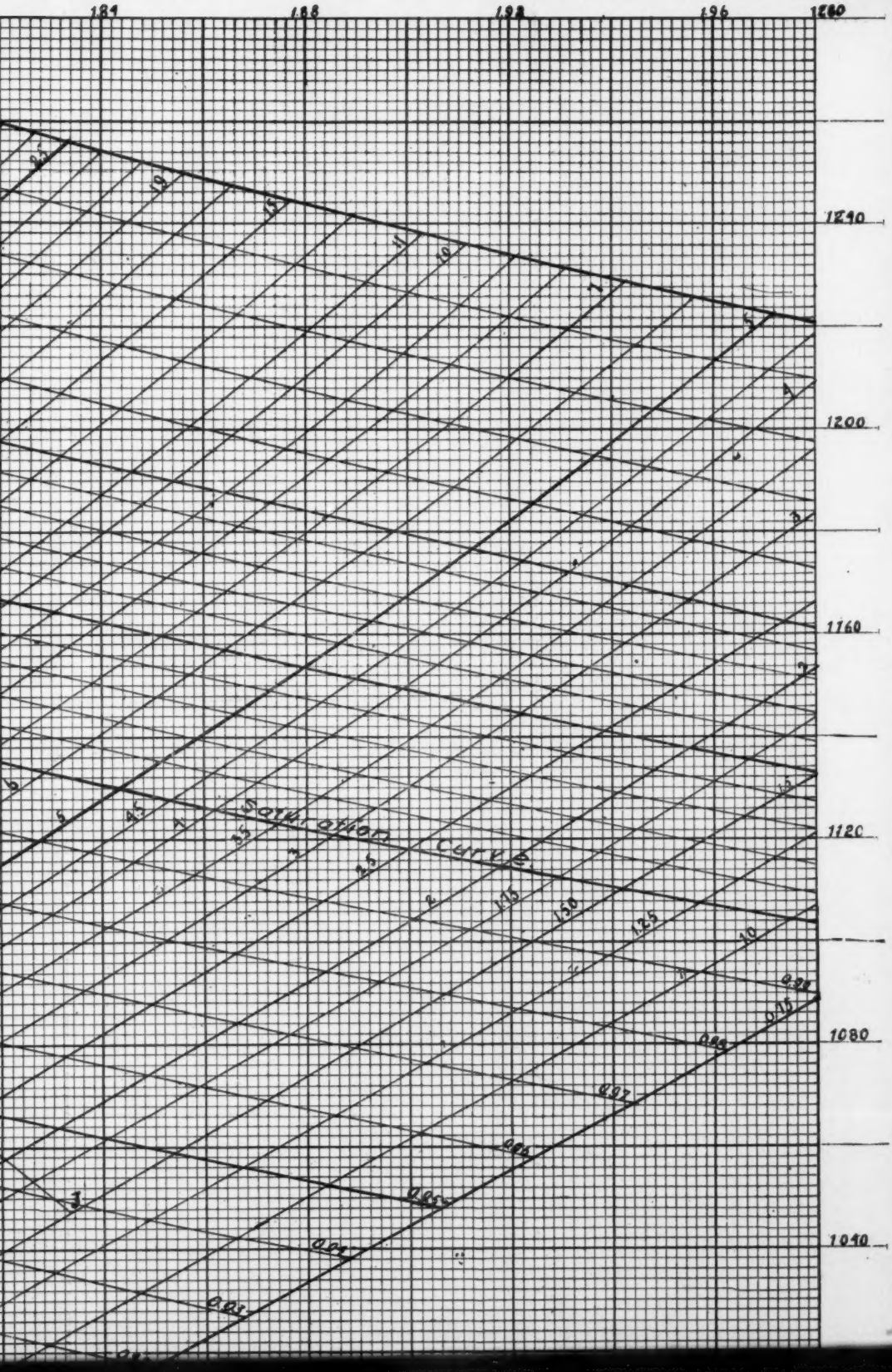
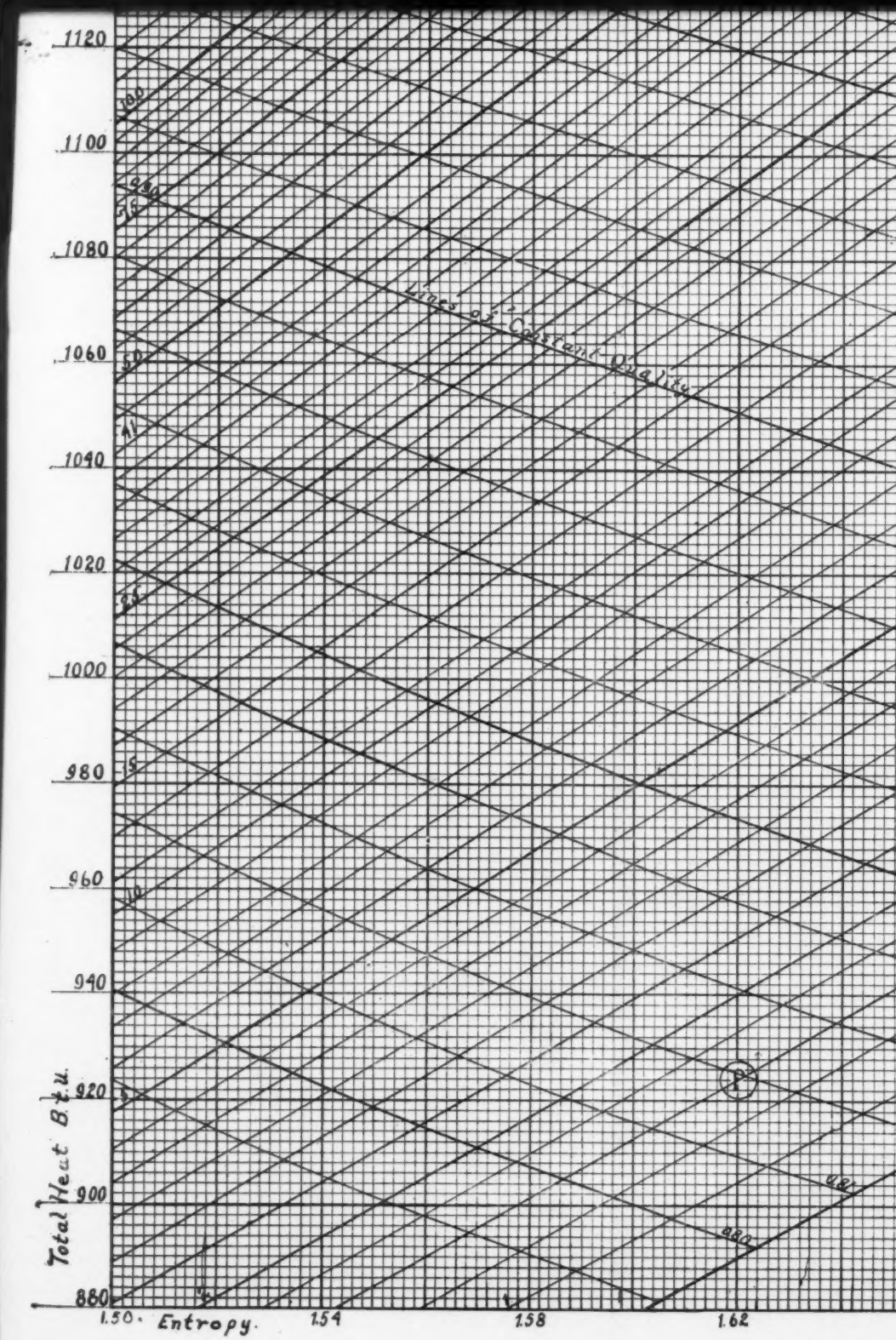
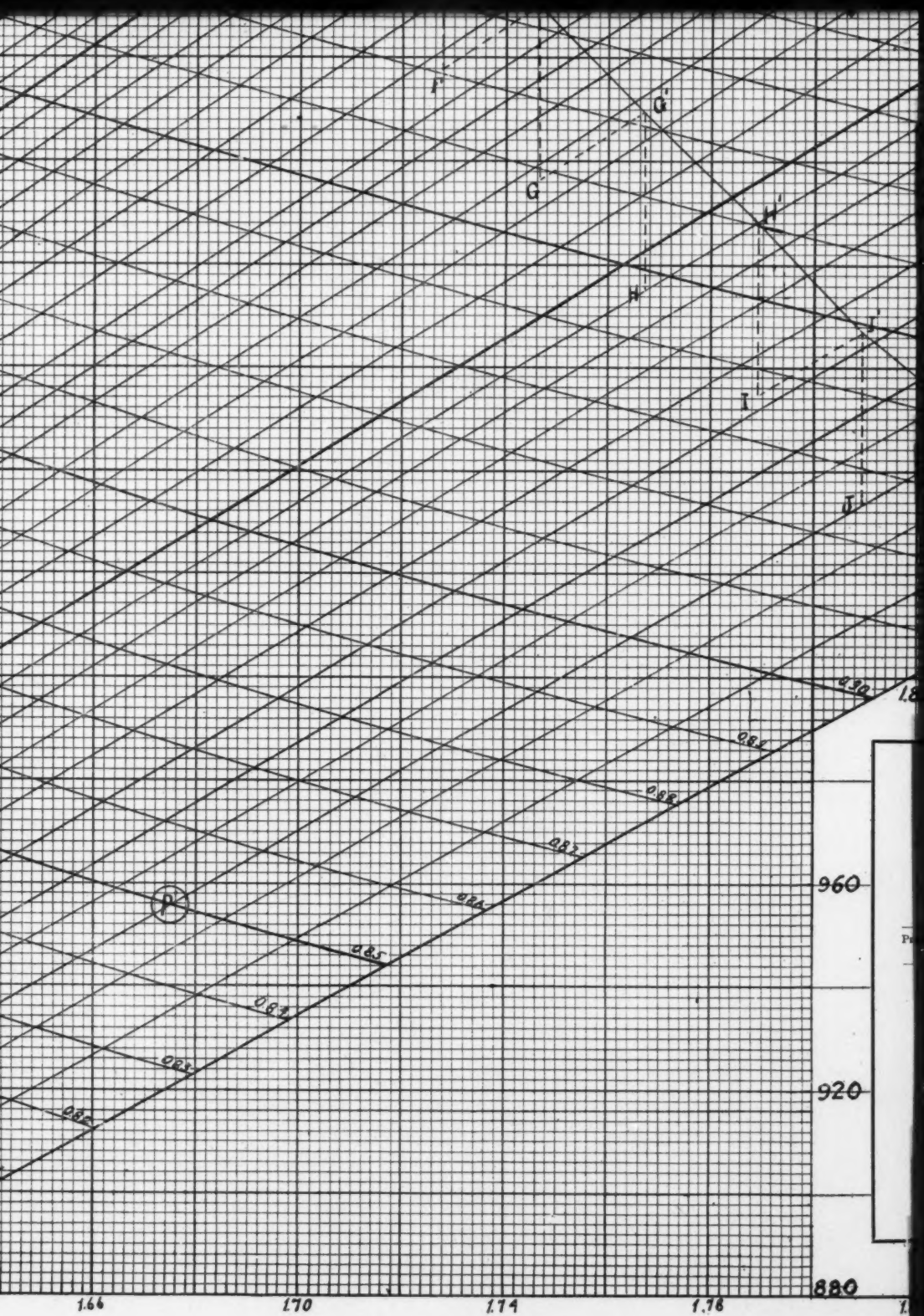
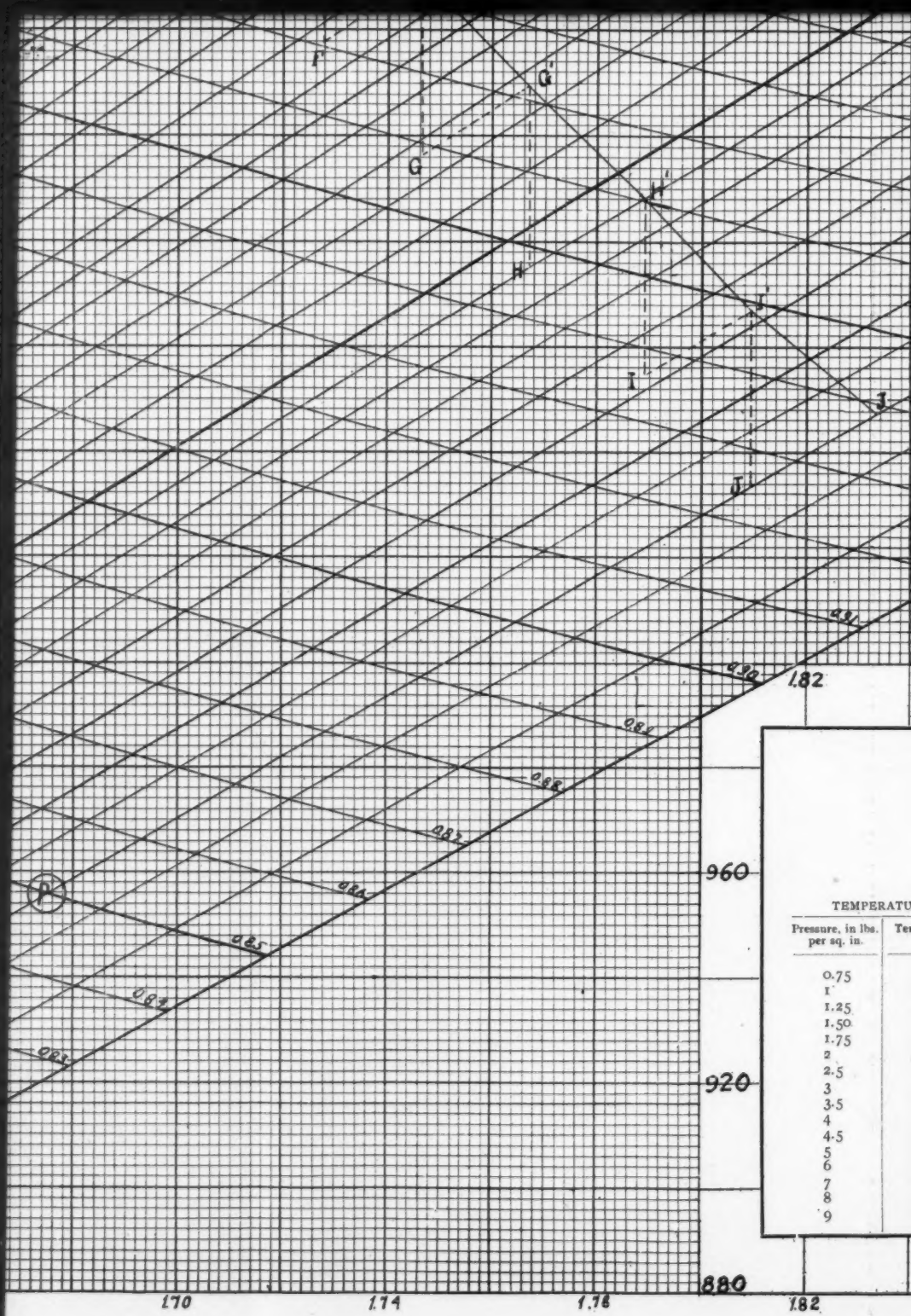

184
188
192
196
1200
1240
1200
1160
1120
1080
1040
Saturation Curve.


1120
1100
1080
1060
1040
1020
1000
980
960
940
920
900
880
Total Heat B.t.u.
1.50
Entropy.
1.54
1.58
1.62
Lines of Constant Quality
100
90
80
70
60
50
40
30
20
10
P²
920
880


F
G
G'
H
H'
I
I'
J
J'
0.90
0.88
0.86
0.84
0.82
0.80
0.85
0.84
0.83
0.82
960
920
880
1.66
1.70
1.74
1.78



TEMPERATU

Pressure, in lbs. per sq. in.	Te
0.75	
1	
1.25	
1.50	
1.75	
2	
2.5	
3	
3·5	
4	
4·5	
5	
6	
7	
8	
9	

SCHOOL OF MARINE ENGINEERING, U. S. NAVAL ACADEMY,
ANNAPOLIS, MD., March 1, 1911.

TOTAL HEAT-ENTROPY DIAGRAM.

Ordinates are total heats; scale 1 inch = 20 B.T.U.
Abscissae are entropies; scale 1 inch = .02 units.
Data taken from Marks and Davis' Steam Tables.

O. L. COX.

PERATURE OF SATURATED STEAM AT EACH PRESSURE FOR WHICH A CURVE IS DRAWN.

ns.	Temperature, °F.	Pressure, in lbs. per sq. in.	Temperature, °F.	Pressure, in lbs. per sq. in.	Temperature, °F.	Pressure, in lbs. per sq. in.	Temperature, °F.
	92.4	10	193.2	45	274.5	140	353.1
	101.7	11	197.8	50	281	150	358.5
	109.4	13	205.9	55	287.1	160	363.6
	115.8	15	213	60	292.7	170	368.5
	121.3	17	219.4	65	298	180	373.1
	126.2	19	225.2	70	302.9	190	377.6
	134.5	21	230.6	75	307.6	200	381.9
	141.5	23	235.5	80	312	210	386
	147.6	25	240.1	85	316.3	220	389.9
	153	27	244.4	90	320.3	235	395.6
	157.9	29	248.4	95	324.1	250	401.1
	162.3	32	254.1	100	327.8	265	406.2
	170	35	259.3	110	334.8	280	411.2
	176.9	38	264.2	120	341.3	300	417.5
	182.9	41	268.7	130	347.4	340	429.1
	188.3						

VALUES OF THE REHEAT FACTOR IN STEAM-TURBINE DESIGN.

A SHORT METHOD FOR DRAWING THE EXPANSION LINE.

By Edgar Buckingham, Associate.

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1. Steady Flow of Steam; the Mollier Diagram.

Let steam be flowing steadily through a channel from a pressure p_1 to a pressure p_2. Let the flow be so rapid or the channel so well protected that the conduction and radiation losses from the steam are negligible as is the case with steam-flow through a large turbine. Let $M=T+W$ be the mechanical energy developed, per pound of steam, between entrance and exit, T being the increase of kinetic energy of the steam, if its velocity has changed, and W the work delivered outside the channel by any motor-forming part of the channel. The quantities M, T, and W are to be expressed in British thermal units (B.T.U.) per pound of steam. Then the first law of thermodynamics gives us

$$M=H_1-H_2. \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where H_1 and H_2 are the values of the total heat of the steam at entrance and exit, H being defined by the equation $H=E+pv$ in which E is the specific internal energy, and pv is reduced, like T and W, to B.T.U.

If no mechanical energy is produced, as in the case of simple wire-drawing into a diffuser, $M=0$ and $H_2=H_1$. *i.e.*, the total heat does not change. Since the total heat of dry saturated steam decreases with the pressure, wire-drawing tends to dry or superheat,—a familiar fact on which the action of the "throttling calorimeter" is based.

If the channel consists of the casing of a steam turbine, the

greater the internal dissipation due to windage, viscosity or eddy currents in blades or nozzles, and wire-drawn leakage past blade-tips or bushings, the greater is the waste of mechanical energy, the greater the resulting reheat which increases H_2 and so decreases the heat-drop $(H_1—H_2)$, and the less the value of M, whether in the form of shaft work delivered or kinetic energy in the exhaust.

In such a dissipative expansion, the reheat causes the same increase of the entropy φ as would an equal amount of heat added, at the same steam temperature, from without. The expansion, though adiabatic, is not isentropic and not reversible. An adiabatic expansion free from the passive resistances which cause dissipation and reheat would be isentropic and, in the thermodynamic sense, reversible. It would also evidently develop the largest possible value of M for adiabatic expansion from the given initial state to the given final pressure. Hence the ideal process for developing the maximum attainable mechanical energy by adiabatic expansion through a steam turbine is an isentropic expansion.

On the H, φ, or Mollier diagram, drawn with H as ordinate, an isentropic expansion is represented by a vertical line, and the value $(H_1—H_3)$ of the maximum heat-drop available for conversion into mechanical energy is equal to the length of this line from the initial state to the final pressure. Any real expansion through a turbine is partially dissipative and forms an intermediate case between isentropic expansion (φ=const.) represented by a vertical line, and complete wire-drawing (H=const.) represented by a horizontal line. The reheat decreases the heat-drop and increases the entropy, and the final state lies higher than the ideal final state and to the right of it. Hence the expansion line AB (Fig. I) will slope downward to the right of the isentropic line AC. The efficiency ε of the process, in converting available heat into mechanical energy, will evidently be

$$\varepsilon = \frac{H_1 - H_2}{H_1 - H_3} = \frac{AD}{AC} \quad . \quad . \quad . \quad . \quad . \quad (2)$$

2. The Expansion Line; Stage Efficiency.

The true expansion line will never be a smooth curve like *AB*, for the steam does not expand uniformly through the turbine. But the state of the steam at a number of similarly placed points along the turbine, *e.g.*, at entrance to the nozzles of the various stages of a turbine of the Curtis, Rateau, or Zoelly type, will be represented by points on a smooth curve. The greater the number of stages and the less the heat-drop in each, the nearer will the true expansion line, with all its details, come to coinciding with a smooth curve similar to *AB* (Fig. I). We shall, for conciseness, speak of this smooth

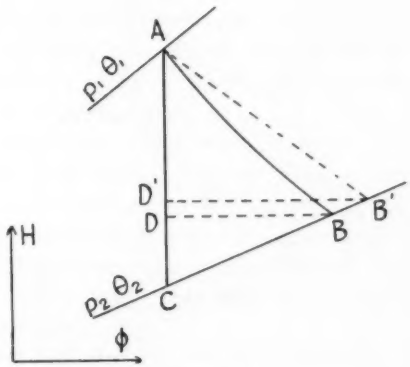


Fig. I.

curve, which is approached as the number of stages between the two terminal pressures increases indefinitely as "the expansion line." It is necessary for the designer to know the approximate form of this expansion line in order that, by reading from the H, φ chart the values of the dryness factor or the superheat, he may find correctly the specific volumes and thence the cross sections needed to pass the required amount of steam at the designed steam speeds, to which the blade speeds and the angles have been adjusted. The form of the line must also be known for determining the pressure limits of a number of intermediate stages, if it is desired to

distribute the work in any prescribed manner among a number of stages of given efficiencies.

By the "stage efficiency" of a given stage shall be meant the value of the quantity ϵ defined by equation (2) when p_1 and p_2 are the terminal pressures of that stage. Let us consider a single stage working through a small range of pressure, p_1 to p_2 with the efficiency ϵ. If A and B (Fig. II) are the initial and final states of the steam at entrance to and exit from this stage, we have $AD/AC = \epsilon$ and

$$\tan b = \frac{\epsilon}{1-\epsilon} \tan a. \quad . \quad . \quad . \quad . \quad . \quad (3)$$

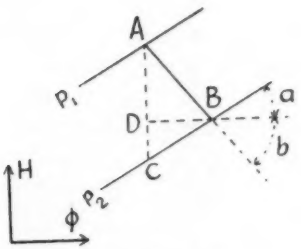


Fig. II.

If the stage efficiency is given, we can thus construct the expansion line by a step-by-step process. From the initial point A we measure a short distance AC downward, and divide it at D so that $AD/AC = \epsilon$; AB will then be an element of the desired expansion line. From B we proceed similarly, and so on down to the final pressure. This construction is applicable at any point of the H, φ plane and not only to the saturation field where the isopiestics are straight lines. The value of ϵ is not required to be constant, but only to be given. The smaller the steps the more accurate the construction if the graphical work is exact.

The problem is often presented of arranging, between two given limiting pressures, a number of similar stages which may be treated as having the same stage efficiency if so de-

signed that the heat-drop shall be the same in each, the intermediate pressures being not given, but to be determined so that the heat-drop and the work on the rotor *shall* be uniformly distributed among the stages, the difference of kinetic energy of the steam between similarly situated points in the turbine being negligible. The solution of this problem by direct graphical construction of the expansion line may involve a considerable expenditure of time if the number of stages is large. Moreover, in making preliminary sketch designs we often want the position of the point B without caring for the precise form of the expansion line. We need, therefore, a quick method of finding the point B (Fig. I) and one or two points on the expansion line between A and B. If this has been done, the expansion line, which is never sharply curved, may be sketched in on an H, φ chart* with sufficient accuracy for all the needs of the designer.

If there are to be n stages, we then divide AD (Fig. I) into n equal parts and draw horizontal lines to intersect the expansion line. These points of intersection fix the pressure limits needed in order that all the stages shall have the same heat-drop. The steam quality may also be read off, and the specific volumes found at entrance to the various stages and at exit from the last stage.

3. Combined Efficiency of Similar Stages in Series; the Reheat Factor.

It may easily be shown that the slope of an isopiestic at any point is given by the equation

$$\left(\frac{dH}{d\varphi}\right)_p = \theta, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

where θ is the absolute temperature. For wet steam, *i.e.*, for all points within the saturation field, the temperature is fixed by the pressure, so that θ is a constant along each isopies-

* Such a chart is contained in the "Steam Tables and Diagrams" of Marks and Davis, Longmans, Green & Co., 1909, from which all the steam data used in this paper have been taken.

tic and the isopiestics are therefore by equation (4), straight lines, increasing in slope with increasing pressure and temperature, and having the familar fan-shaped arrangement.

It is therefore evident from figure II, that the expansion line for a series of stages of constant stage efficiency will slope downward more sharply at the start than later, and will be concave to the right, as shown by AB in figure I. It will cross every intermediate isopiestic with such a slope as to satisfy equation (3), and will meet the final isopiestic p_2 (Fig. I) at the same angle as a line AB' drawn to make $AD'/AC=\varepsilon$, but at a point B which is lower than B'.

The combined or series efficiency ε_c of a number of similar stages is thus greater than their separate stage efficiency ε in the ratio

$$\frac{\varepsilon_c}{\varepsilon}=\frac{AD}{AD'}=R. \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

This ratio R is known as the "reheat factor." If its value can be found a priori, the step-by-step construction of the expansion line may be dispensed with and the point B found at once from the given stage efficiency, the initial state A, and the final pressure p_2. Similar points found for one or two intermediate pressures then enable us to draw the expansion line sufficiently accurately. The accuracy of the whole process of designing is limited by the accuracy with which the stage efficiency can be predicted, and this is not better than one per cent. Hence if the approximations used in designing are such as to introduce errors of the order of only ½ of one per cent. or less, they are close enough, and time would be wasted in striving for a precision not warranted by the fundamental datum, the stage efficiency.

A general solution of the problem of determining the reheat factor a priori is not possible at present, but for wet steam a solution is given in the following sections, the mathematical work being condensed to a mere outline of the process used. The method followed is first, to deduce an expression for R from an approximate assumption regarding the distribution of

the isopiestics within the saturation field; second, to derive a convenient practical method by which computation is nearly eliminated; and, third, to show that this method gives sufficiently accurate results, by making numerical comparisons with a more exact expression for R based on a much more accurate assumption regarding the distribution of the isopiestics.

4. The Reheat Factor for Wet Steam; First Approximation.

Examination of the chart shows that the isopiestics, when produced do not meet in a single point. But the intersections are so far to the left and the total angle between any two isopiestics needed in practice, say 1 pound and 300 pounds absolute, is so small that no large error is incurred by treating them as if they did all radiate from a single point O (Fig. III). Our first approximation will be based on this assumption.

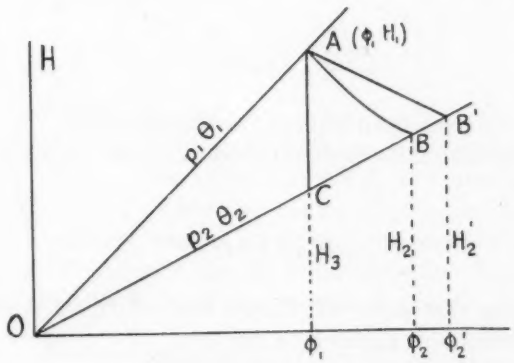


Fig. III.

Let H and φ be measured from O as origin. In our results, only differences of H will appear, so that the fact that the charts actually used do not satisfy this condition does not affect the correctness of the conclusions. Let a and b have the same meaning as in equation (3) and figure II.

Then at any point of the expansion line we have for the slope of the line

$$\kappa \tan b = -\frac{dH}{d\varphi}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

The value of the constant κ depends on the scales of H and φ; it would be unity if one entropy unit were represented by the same length as a difference of 1 B.T.U. in H.

For the isopiestic passing through this same point we have

$$\kappa \tan a = \left(\frac{dH}{d\varphi}\right)_p = \frac{H}{\varphi}, \quad . \quad . \quad . \quad . \quad . \quad (7)$$

for $\left(\frac{dH}{d\varphi}\right)_p$ is constant along the isopiestic, and the isopiestic passes through the origin O from which H and φ are measured. Substituting in equation (3) from (6) and (7), we get

$$\frac{dH}{d\varphi} + \frac{\epsilon}{1-\epsilon}\frac{H}{\varphi} = 0. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

as the differential equation of the expansion line.

Integrating this equation between any two points A and B, we get

$$H_2 \varphi_2^{\frac{\epsilon}{1-\epsilon}} = H_1 \varphi_1^{\frac{\epsilon}{1-\epsilon}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

Noting that $\varphi_2/\varphi_1 = H_2/H_3$ and that $H_3/H_1 = \theta_3/\theta_1 = \theta_2/\theta_1$ we may reduce this equation to the form

$$H_2 = H_1\left(\frac{\theta_2}{\theta_1}\right)^{\epsilon}. \quad . \quad . \quad . \quad . \quad . \quad (10)$$

an equation for the expansion line in terms of temperature and total heat. The heat-drop between A and B is therefore

$$H_1 - H_2 = H_1\left[1 - \left(\frac{\theta_2}{\theta_1}\right)^{\epsilon}\right]. \quad . \quad . \quad . \quad . \quad (11)$$

If the combined efficiency were the same as the stage efficiency the heat-drop from A to B' would be

$$H_1-H'_2=\varepsilon\,(H_1-H_3)=\varepsilon H_1\left(1-\frac{\theta_2}{\theta_1}\right). \quad . \quad . \quad (12)$$

Hence the value of the reheat factor is given by the equation

$$R=\frac{1-\left(\frac{\theta_2}{\theta_1}\right)^{\varepsilon}}{\varepsilon\left[1-\frac{\theta_2}{\theta_1}\right]}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (A)$$

5. Practical Rule for Finding the Value of R.

Values of R were computed by equation (A) for a number of cases and it was noticed that if p_1, p_2 and p_3 are any three pressures, and R_{12}, R_{23}, and R_{13} are the reheat factors for expansion between these pressures with the same stage efficiency, the relation

$$(R_{12}-1)+(R_{23}-1)=(R_{13}-1). \quad . \quad . \quad . \quad . \quad (B)$$

was almost exactly satisfied.

It was next found that for expansion, with constant stage efficiency, from a given initial pressure p_1 and temperature θ, to any lower temperature and pressure p_2 θ_2, the values of R or $R-1$ plotted against $\log \theta_2$ lay on a straight line, the slope of which depended on the stage efficiency. It was then further found that the empirical equation

$$R-1=1.207\ (0.975-\varepsilon)\ (\log \theta_1-\log \theta_2). \quad . \quad . \quad (C)$$

represented the values of R obtained from equation (A) as closely as could be desired for practical purposes, *i.e.*, to a small fraction of one per cent.

The initial pressure p_1 was now taken at 350 pounds, for which $\log_{10} \theta_1=2.95012$, and values of R were computed by equation (C) for expansion from p_1 down to eighteen well distributed lower pressures, the lowest being 0.5 pounds, and

with ϵ=0.1, 0.2...etc., ... 0.8. These values, together with the pressures to which expansion takes place, are given in table 1.

Table 1.

p_1	ϵ=0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8
300	1.0075	1.0066	1.0058	1.0049	1.0041	1.0032	1.0023	1.0015
250	1.0161	1.0143	1.0124	1.0106	1.0088	1.0069	1.0051	1.0032
200	1.0265	1.0235	1.0205	1.0174	1.0144	1.0114	1.0083	1.0053
150	1.0394	1.0349	1.0304	1.0259	1.0214	1.0169	1.0124	1.0079
100	1.0569	1.0504	1.0439	1.0374	1.0309	1.0244	1.0179	1.0114
70	1.0717	1.0635	1.0553	1.0471	1.0389	1.0307	1.0225	1.0143
40	1.0936	1.0829	1.0722	1.0615	1.0508	1.0401	1.0294	1.0187
30	1.1047	1.0927	1.0808	1.0688	1.0568	1.0449	1.0329	1.0209
20	1.1191	1.1055	1.0919	1.0783	1.0646	1.0510	1.0374	1.0238
15	1.1292	1.1144	1.0997	1.0849	1.0702	1.0554	1.0406	1.0258
10	1.1429	1.1265	1.1102	1.0939	1.0775	1.0612	1.0449	1.0285
7	1.1545	1.1369	1.1192	1.1016	1.0839	1.0663	1.0486	1.0309
5	1.1651	1.1462	1.1274	1.1085	1.0896	1.0708	1.0519	1.0330
3	1.1808	1.1601	1.1395	1.1188	1.0981	1.0775	1.0568	1.0361
2	1.1926	1.1706	1.1486	1.1266	1.1045	1.0825	1.0605	1.0385
1.5	1.2012	1.1782	1.1552	1.1322	1.1092	1.0862	1.0632	1.0402
1.0	1.2120	1.1878	1.1636	1.1394	1.1151	1.0909	1.0667	1.0424
0.5	1.2309	1.2045	1.1781	1.1417	1.1253	1.0989	1.0725	1.0460

The values of $(R-1)$ for ϵ=0.1 are plotted on plate 1 as ordinate against p_2 as abscissa, so that this curve represents the straight line.

$$R-1=1.207(0.975-0.1)(\log p'_1-\log \theta_2).$$

with the abscissa changed from log θ_2 to p_2, which is usually given directly and is therefore more convenient to work with than θ_2.

Having this curve, and bearing equation (B) in mind, we now have, for finding the reheat factor applicable to expansion with stage efficiency ϵ between any two pressures p_1 and p_2 the following

RULE: *Head from the curve the values of $(R-1)$ at the given pressures; their difference is the required value of $(R-1)$ for ϵ=0.1. For any other value of ϵ, multiply by the factor* $\frac{0.975-\epsilon}{0.875}$.

The process of finding R thus consists in making two readings from a curve and multiplying their difference by a factor which may be tabulated once for all, so that the whole operation takes little time. For example:

Let it be required to find the reheat factor for adiabatic expansion of wet steam from 240 to 60 pounds with a stage efficiency $\varepsilon=0.52$. Reading from the curve we find: at $p=60$, $R-1=0.078$; and at $p=240$, $R-1=0.018$. Hence from 240 to 60 we have $R-1=0.060$. With $\varepsilon=0.52$ we therefore have

$$R-1=0.060\times\frac{0.975-0.52}{0.875} \text{ and } R=1.031.$$

This process might be still further simplified by drawing on a single sheet the curves for $\varepsilon=0.1$, 0.2, 0.3..., etc., and interpolating for the given value of ε, thus avoiding the multiplication. The values in table I are given so that anyone who wishes to do so may plot such a set of curves for his own use.

6. Second Approximation for R.

Equation (A) was deduced from an assumption known to be inexact. It remains to be shown that the resulting error in R is negligible, and this will be done by deducing a more exact expression for R, and showing that the result is sensibly the same as that obtained by equation (A) or by the use of the rule just given.

If the isopiestics all met in a single point the values of Z, where Z is the value of H at the intersection of any isopiestic with some fixed isentropic, would evidently satisfy the equation

$$\theta=l+mZ, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

in which $l=o$ if H is counted from the origin O. This equation is therefore one expression for the assumption which underlies the deduction of equation (A). By readings from the chart at three widely separated isentropics it was found that the equation

$$\log \theta=A+BZ. \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

represented the readings about as accurately as they could be made. An expression for R based on equation (14) will therefore certainly give more nearly the true values than equation (A) based on equation (13). Equation (14) which may also be written in the form

$$H=C+D\log\theta. \quad . \quad . \quad . \quad . \quad . \quad (15)$$

giving the variation of total heat with absolute temperature during isentropic expansion, is an approximate relation which I have not seen given elsewhere and which might prove useful in other problems than the present one.

By methods somewhat similar to those used in obtaining equation (A) we may now deduce, for the expansion line, the differential equation

$$\frac{dH}{dZ}=\varepsilon[1+B(H-Z)], \quad . \quad . \quad . \quad . \quad (16)$$

in which H is the ordinate of any point P on the expansion line, and Z the corresponding ordinate of the point where the isopiestic through P cuts the fixed isentropic to which equation (14) refers. If H_1 Z_1 θ_1 refer to the initial state of the steam at the pressure p_1, and H_2 Z_2 θ_2 to the final state after adiabatic expansion to the pressure p_2 with stage efficiency ε, we get, after integrating equation (16),

$$R=\frac{Z_1-Z_2+K\left[1-\left(\frac{\theta_2}{\theta_1}\right)^{\epsilon}\right]}{\varepsilon\left[Z_1-Z_2+L\left(1-\frac{\theta_2}{\theta_1}\right)\right]}. \quad . \quad . \quad . \quad . \quad (D)$$

where

$$L=H_1-Z_1. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (D_1)$$

and

$$K=L-\frac{1-\varepsilon}{\varepsilon B}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (D_2)$$

The readings of Z for the isentropic $\epsilon=1.384$ were plotted against $\log\theta$ and a straight line drawn through the points; the values of Z needed for use in equations (D) were read from this plot. The slope of the line gave the value $1/B=723$. for use in (D_2). The values of H_1 for use in equation (D_1)

were read from the chart. It was found that if H_1 was taken from the tables and used in connection with values of Z and B obtained from readings on the chart, considerable inconsistencies resulted, showing that the chart while self-consistent and sufficiently accurate for ordinary purposes was not an exact representation of the tables, owing, doubtless, to difficulties in printing. This need not, however, occasion any lack of confidence in the chart, for it will be found, on making a few computations by equation (D) that this is a severe test. The value of R given by equation (D) is greatly influenced by small errors in the data, and the equation is not suitable except for careful work, quite aside from its inconvenient complexity.

The use of this cumbersome equation proves, fortunately, to be quite unnecessary. A number of test computations covering the part of the saturation field which is of practical importance, showed that equation (D) gave sensibly the same results as were obtained by using the curve according to the rule given in section 5. In many cases the values of R were also computed by equation (A). On the whole, the values obtained from the curve agreed with those from equation (D) rather better than did those obtained from equation (A). A summary of the results of these tests is given in table 2.

Table 2.

Value of the change of position of B expressed as percentage of $(H_1—H_2)$

$\frac{p_1}{p_2}$	p_1	x_1	ε=0.1	ε=0.2	ε=0.4	ε=0.6	ε=0.7
12.7	277.4	0.981		+0.03	—0.02	—0.02	
45.5	277.4	0.981		+0.04	—0.09	—0 10	
185.	277.4	0.981		—0.03	—0.20	—0.22	
44.	66.2	1.00		—0.06	—0.09	—0.10	
33.7	21.9	0.99		+0.04	—0.04	+0.05	
33.7	21.9	0.88		+0.04	—0.06	+0.07	
33.7	21.9	0.77		+0.06	+0.07	—0.08	
20.	20.	0.99	+0.04		—0.01		—0.03
20.	20.	0 84	+0.04		+0.04		—0.02
20.	20.	0.79	+0.04		—0.02		—0.01
20.	120.	0.99	+0.07	+0.05	+0.03		+0.04
20.	120.	0.84	+0.08	+0.04	+0.03		+0.04
300.	300.	1.00		—0.01	—0.19	—0.25	
110.	110.	1.00		—0.06	—0.13	—0.10	
35.	35.	1.00		+0.07	+0.01	—0.01	
8.	8.	1.00		+0.03	+0.01	+0.02	

The first three columns give the pressure ratio, the initial pressure and the initial dryness factor x_1, and it will be seen that the tests are sufficiently varied. The object in using the reheat factor is to determine the position of a point B with an error which shall be as small as possible compared with the isentropic heat-drop $(H_1—H_3)$. This discrepancy between the results of the two methods of computing R is found, in percentage of $(H_1—H_3)$, by multiplying the difference of the two values of R (R being always nearly unity) by the value of ε. Accordingly, the values in the table may be regarded as the changes in the vertical height of the point B, expressed in per cent. of $(H_1—H_3)$, caused by changing the method of computing R.

On account of errors in reading H, Z, and $(R—1)$ from the curves, differences of less than 0.1 have little or no significance except when p_1/p_2 is large. It will be seen that the discrepancy between the two methods exceeds 0.1 per cent. only when the pressure ratio is very large and that it in no case exceeds 0.25 per cent. Since ¼ of one per cent. is negligible for practical purposes, we may conclude that since the use of equation (D) certainly gives more accurate values than the rule, and since the difference of the resulting values is negligible, the rule itself is accurate enough. It follows that values of R obtained by the rule of section 5 may be relied upon for wet steam as being sufficiently accurate for the purposes of the designer.

7. General Remarks.

The foregoing methods are valid only when the isopiestics are straight lines; they therefore fail for superheated steam. The curvature of the isopiestics in the superheat field depends on the specific heat of superheated steam at constant pressure, and even if this were exactly known and expressible in simple mathematical form, the development of a general differential equation for the expansion line would be a difficult matter and the task of integrating it would very possibly present in-

superable mathematical difficulties. But without attempting the general solution a few pertinent remarks may be made.

(A) Consideration of the H, φ chart shows that an isentropic expansion between two given isopiestics involves a greater change in temperature for superheated than for wet steam, as may also be seen from the p, v chart. A dissipative expansion line starting at a point in the superheat field will therefore fall more sharply at first than if it started at the same pressure within the saturation field, and the reheat factor for a given pair of terminal pressures and given stage efficiency will be greater than for wet steam between same pressures. Hence if we use the value of R computed for these pressures for wet steam, we shall underestimate the actual heat-drop and combined efficiency and be on the safe side, as regards the economy expected. The small error committed will be in the direction of raising the expansion line.

(B) Up to 100 degrees F. superheat the curvature of the isopiestics is so small that the error can, at most, not be important if we simply use the value of R for wet steam between the same pressure limits.

(C) The value of the quantity of $(R—1)$ decreases with the pressure ratio p_1/p_2 and decreases as the stage efficiency increases. Examination of the H, φ chart in connection with table 1, shows that unless the efficiency is very poor, with superheats up to 150 degrees or 200 degrees F., which are unusual as yet, the expansion ratio and therefore $(R—1)$ cannot be large for expansion entirely within the superheat field, so that $(R—1)$ cannot be subject to a large error. With poor efficiency the reheat might be enough to keep the steam dry a long way down in the turbine, but such a case is not of great commercial interest and does not demand close designing.

(D) A high initial superheat has not yet been much used in the Parsons type of turbine, so that the superheat field is of interest mainly to the designer of turbines in which the first few stages, at least, are of the impulse type, not requiring fine clearances. In such a turbine, the first one or two stages will usually have rather large pressure ratios and will get rid of the

superheat, so that the remaining stages are working with wet steam. It is not a serious matter to treat one or two stages separately, by determining the state-points graphically, and it is usually necessary to do so for at least one stage. For the first stage is usually not identical with the later stages and has not the same stage efficiency, so that an expansion line drawn for constant stage efficiency could not fit the facts in any event.

(E) The whole idea of the reheat factor and of its use in drawing the expansion line, arises from the fact that the combined efficiency of a number of similar stages in series is greater than the efficiency of each stage separately. The method developed for finding the reheat factor was based on the assumption that the heat-drop in each stage was small, for it consisted in developing and integrating a *differential* equation for the expansion line. For a few stages with very high steam velocities, the results would not be exact, but the whole matter is also, in that case, of no importance; for the graphical construction for a few stages is a simple operation and involves no great expenditure of time.

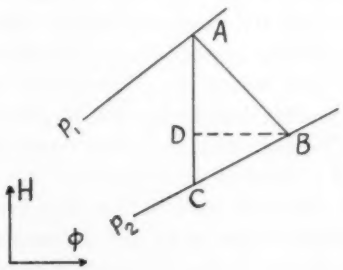


Fig. IV.

(F) The notion of a reheat factor, as the term has been used in this paper, is not applicable to a single stage. Let p_1 p_2 be the limiting pressures of a given single stage, and let A (Fig. IV) be the initial state. Then if B is the final state,

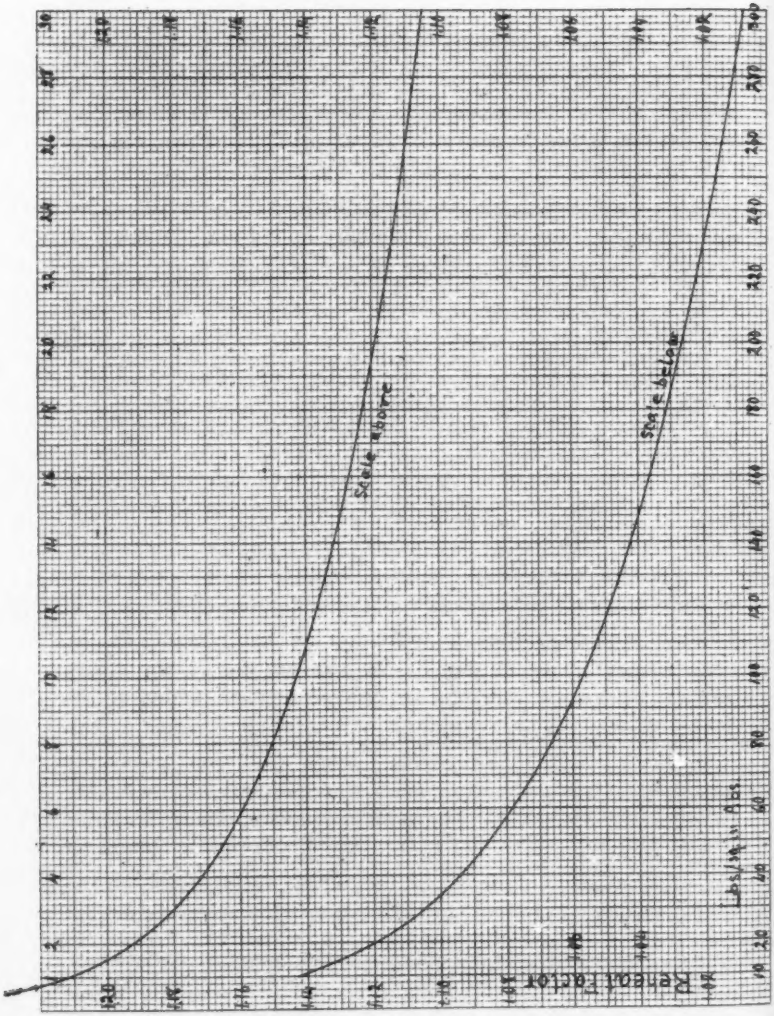
$$\frac{H_1-H_2}{H_2-H_3}=\frac{AD}{AC}$$

is the ratio of the mechanical energy produced to the maximum possible in adiabatic expansion. If, as may be assumed in an impulse stage,* the change of kinetic energy is negligible, $AD/AC=\varepsilon$ is the efficiency of the stage. This is true regardless of whether the isopiestics are straight or curved, *i.e.*, whether the steam is wet or superheated, so long as B is at p_2, AC is vertical, and DB is horizontal. If the net efficiency of the stage is given beforehand, B is thus fixed by the initial state A and the final pressure p_2, and there is no question of a reheat factor.

But the efficiency ε may have been obtained in either of two ways. If it was obtained by experiments on a stage similar to the one in question and working under similar conditions, there is nothing more to be said. But if it was computed from a velocity diagram drawn with allowances for velocity losses in nozzles and blades, the case may be a trifle different in theory though hardly in practice.

Let A, and therefore C, be given, and let D be so placed as to make $AD/AC=\varepsilon$, where ε is the computed efficiency. It now becomes a question of when the various elements of the total dissipation occur and of the temperature of the steam when the various elements of the reheat are added to it. In an impulse stage, in general, only a small fraction of the dissipation occurs in the nozzles: hence the reheat is almost all added to the steam after the lowest pressure and temperature have been reached. The actual expansion line is therefore nearly coincident with the broken line ACB, which would be the real expansion line if *all* the reheat were added at the low pressure. If, therefore, the efficiency was computed on

*It is not assumed that the flow through either nozzles or blades is frictionless, and nothing is assumed regarding the residual velocity of the steam on leaving the last moving blades of the stage, which is sometimes partially available at entrance to the next nozzles, but is more often entirely thrown away and dissipated in eddy currents. The sole assumption is that when the steam finally leaves the stage by entering the next nozzles, its velocity is so nearly the same as on entering the nozzles of the given stage, that the *difference* in kinetic energy is negligible.


Reheat Factor
Lbs./sq. in. Abs.
Sat. Vapor
Sat. Water

this basis, while in reality a little of the reheat is added at a higher pressure and temperature in the nozzles, the efficiency thus computed will be a trifle too small and should be multiplied by a factor analogous to the reheat factor, which, in effect, is introduced to allow for the fact that in a series of stages the reheat is distributed instead of being concentrated at the end of the last stage. The improvement in accuracy due to the use of such a factor would, however, be quite illusory until our experimental knowledge is sufficient to permit of a far more accurate a priori computation of the stage efficiency than is at present possible.

To apply a reheat factor computed for the case of continuous expansion through an infinite number of stages to expansion between the same pressure limits through a single stage, would be altogether incorrect.

(G) In conclusion, we may say that so far as present practice is concerned, the fact that the method given in this paper for finding the reheat factor is applicable only to wet steam, is an unimportant restriction on its usefulness. In the designing of turbines with many similar stages, its use may save a great deal of time, for a correct determination of two points beside the starting point will always, even with large expansion ratios, permit of drawing the expansion line so as to give the specific volume of the steam at any pressure within one per cent., which is quite close enough for designing the sectional areas, in view of the unavoidable uncertainties of the stage efficiency.

Washington, March 13, 1911

NOTE.—A more extended form of this paper will appear in the ***Bulletin of the Bureau of Standards***, where those interested may find more complete details.

TEST OF A BABCOCK & WILCOX BOILER WITH LIQUID FUEL.

From the Board's Report.

This test was conducted by a Board composed of Captain C. A. Carr, U. S. N., and Lieutenant Commanders J. K. Robison and John Halligan, Jr., U. S. N.

The Board assembled at the works of The Babcock & Wilcox Company, Bayonne, N. J., about 10 A. M., November 28, 1910. The Board examined the test boiler and its connections for water and steam tightness, the arrangements for measuring water and oil and the apparatus for taking data. All of these were found satisfactory, and the tests were proceeded with at once. In all these tests oil was used as a fuel.

Civilian assistants from the office of the Inspector of Machinery, Bayonne, N. J., were detailed for the purpose of taking data. Another set of observers was furnished by The Babcock & Wilcox Company who took data independently. The data taken by the two sets of observers were compared, so that any errors in reading were corrected at once.

The boiler tested was of the type installed on the U. S. S. *Wyoming* and U. S. S. *Arkansas*, and is the same which was tested June 13 to June 20, 1910, during which tests coal was used as a fuel.* No changes have been made in the design of the boiler except for the removal of the grate, the bricking over of the ash pans and the sides up to the side boxes, and the necessary changes in the furnace front incident to the installation of the oil-burning apparatus.

The following particulars apply to the installation for burning oil:

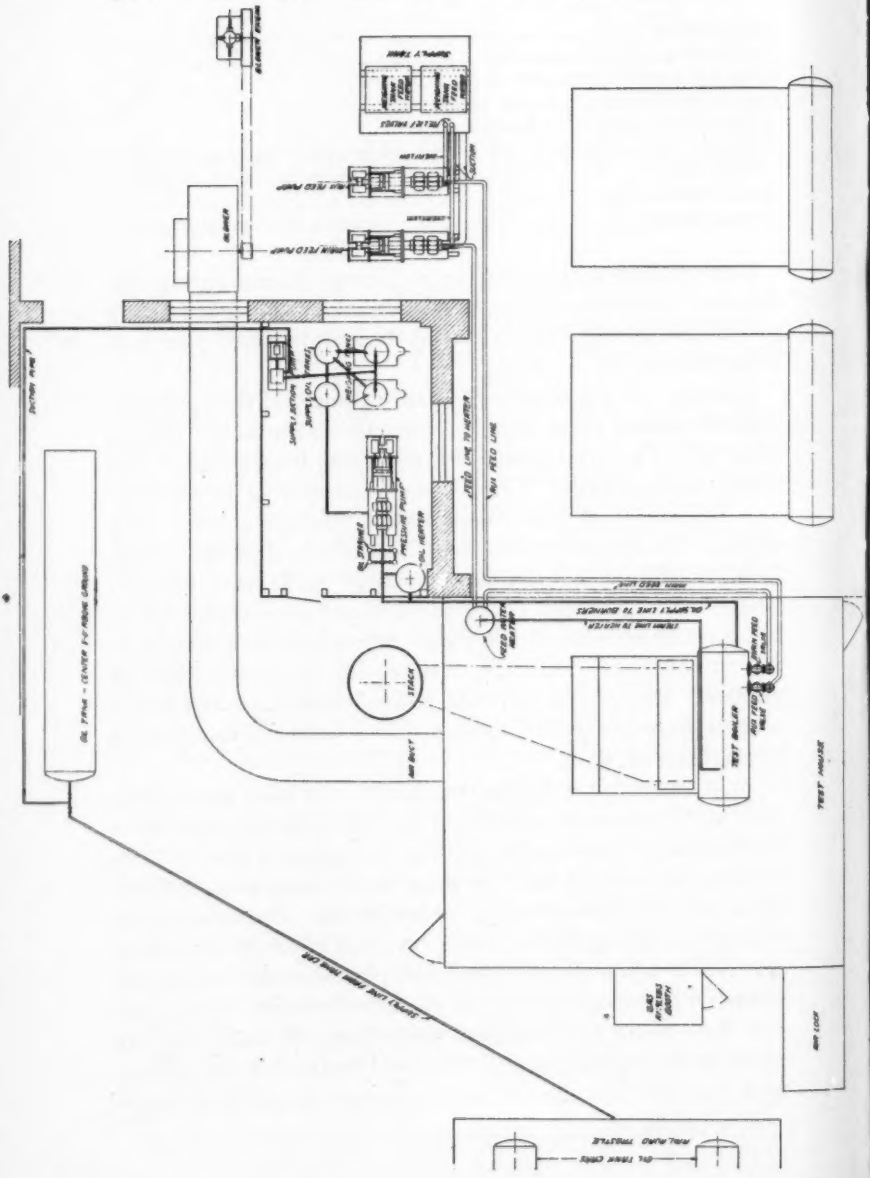
* Reports of these tests appeared in the November, 1910, issue of the JOURNAL.

Type of boiler	Babcock & Wilcox.
Total heating surface, square feet	2,571
Volume of furnace, cubic feet	217
Area of cross section smokepipe, square feet	19.63
Height of smokepipe above furnace, feet	100
Fuel used	Crude oil from Gulf Refining Company.
Kind of draft	Closed fireroom and jet in smokepipe.
Number of burners	11
Type of burner	Peabody Mechanical Atomizer.

For convenience in referring to burners in use during the different tests they are numbered from left to right in each row consecutively, the left-hand burner in the upper row being No. 1.

General Arrangement.—The arrangement of the apparatus used in making these tests is shown on the attached sketch, Sheet B. The arrangement for supplying forced draft is also shown on this sheet. The blower for supplying forced draft was driven by a vertical steam engine through a belt. The air duct discharged at the floor line of the fireroom at the back of the boiler against a vertical baffle wall that deflected the air upward. The main steam pipe connected to the main steam pipe of the power house and by a bleeder to the atmospheric discharge, terminating in a three-branch muffler. The discharge to the power-house main and to the bleeder were controlled by stop valves with extension stems to the fireroom floor.

Feed Heater.—A Reilly feed heater was used for heating the feed. A branch pipe from the main steam pipe led to the feed-water heater for supplying the necessary steam. The pressure of steam in the feed-water heater shell was regulated by a valve in the steam pipe to the heater. This feed heater was in use during all the tests except for about 42 minutes in test No. 2, when the auxiliary feed pipe was used, owing to a temporary derangement of the main feed pump. The auxiliary feed pump took weighed water from the main feed the same as the regular pump. The feed heater was tested before and after the tests under a water pressure of 220 pounds and was found tight.


TEST HOUSE
TEST BOILER
STACK
MAIN FEED LINE
MAIN STEAM LINE
OIL TANK - CENTER FOR FILLING CARLOAD
OIL TANK CHIEF
REAL PUMP TRUSTEE

Steam Jet.—A pipe led from the main steam pipe to a jet in the base of the smokepipe, the amount of opening being regulated by a valve. This jet was used for increasing the draft, as noted in the log of the tests.

Smokepipe.—The smokepipe was of sheet steel, 19.63 square feet area of cross section, and 100 feet in height above the furnace. There was a damper in the smokepipe, which was wide open during all the tests.

The discharges from the surface and bottom-blow valves led into a pipe which led outside of the building and the end of which was open. A slight leak was found at the end of this pipe soon after starting test No. 1, which could not be stopped by setting down on the blow valves, and the end of the pipe was therefore plugged, insuring tightness. It was examined during every succeeding test and no further leaks occurred.

Thermometers were placed in the main feed pipe as near the boiler-feed stop valve as possible, and in the oil-pressure pipe near the burners, and a nitrogen-filled thermometer in the uptake for obtaining the temperature of the escaping gases. Thermometers were also hung in the fireroom and outside the testing room for observing the temperatures of the air.

The draft pressure was taken through a tube in the highest dusting door of the last gas passage. In tests Nos. 4, 5 and 6 it was also taken through a tube in the lowest dusting door of the first gas passage. The air pressure in the fireroom was also measured by an air-pressure gage hung on the bulkhead of the testing room.

Gas Analysis.—An Orsatt apparatus for making analyses of uptake gases was located in a small booth adjacent to the test room. Gas samples were taken from the uptake by means of a ¼-inch pipe leading to the gas-analysis apparatus. Samples of gas were taken at frequent intervals by the chemist of The Babcock & Wilcox Company. The results of the analyses were immediately reported to the company's engineer for his information.

Method of Weighing Oil.—The arrangement of the oil-feeding apparatus is shown on "Sheet B." Two barrels for receiving and weighing oil were mounted on platform scales. These scales were examined before and after the tests for correctness, and were checked during the tests by weighing empty barrels. Oil was run into these barrels for weighing direct from the tank car in which it was received at the works. The flow of oil was assisted by a small pump, the location of which is shown on the sketch. After being weighed the oil was run into one of the two receiving barrels shown. These two barrels were connected at the bottom by a 4-inch pipe, the oil-feed suction being led into the second barrel. This second barrel also received the overflow oil from the relief valves of the oil-pressure pump. From the top of this barrel the height of the oil was measured by a gage, to determine the quantity of oil burned. From the pressure pump the oil passed through strainers to the oil heater. A pressure gage and a thermometer were fitted in this room on the oil-supply pipe for convenience in regulating the temperature and pressure of the oil. The quantity of oil burned could be determined at any instant by measuring the level of oil in the barrel. At half-hour intervals the quantity of oil was checked up.

Method of Weighing and Pumping Water.—The arrangements for weighing feed water consisted of a rectangular feed tank on top of which were located two weighing tanks mounted on platform scales. These scales were tested before and after the tests and the weights of empty tanks also served as a check. The water was run from the city main into the weighing tanks and after being weighed was run into the feed tank. The feed tank was fitted with a gage glass on which the height of water at the start was marked. By regulating the flow of water from the weighing tanks the quantity of water used during any time could be determined. This feed tank had suctions to main and auxiliary feed pumps and also received the overflow from the relief valves of these pumps. The main feed pump discharged through the feed-water heater to the main feed valve on the boiler. All feed piping was where it could be seen, and no leaks took place

during any of the tests. A steam gage and a water gage indicating steam pressure and water level in the boiler were located at the feed pump for assisting in pump regulation and in the maintenance of a constant feed supply. At the end of each half hour the actual amount of water fed to the boiler was checked up.

Quality of Steam.—For observing the quality of the steam generated a Barrus throttling calorimeter was fitted on a branch from the main steam pipe at a point 18 inches from the steam drum. The collecting nozzle was of standard pattern. The calorimeter was calibrated after the tests by taking readings with the pressure both rising and falling with no steam leaving the boiler except through the calorimeter. The calorimeter thermometers and feed-water thermometer were compared with a standard thermometer after the tests, and readings were corrected as necessary. From these standard readings the amount of moisture in the steam was determined from the formula:

$$Q = \frac{.48\,(T - t)}{L} \times 100;$$

in which Q = percentage of moisture; T = calibration reading of the lower thermometer; t = test reading of lower thermometer; L = latent heat of steam at boiler pressure.

Quality of Oil Used.—The oil used was Texas crude furnished by the Gulf Refining Company. Samples of oil were taken direct from the cars, and the characteristics of the oil were determined by a laboratory test made by the chemist at the Navy Yard, Washington, D. C. Oil from car No. 1 was used in tests Nos. 1, 2, 3 and 4. Oil from car No. 2 was used in tests Nos. 5 and 6. The analysis of the oil was as follows:

	Car No. 1.	Car No. 2.
Character of oil	Heavy and viscid.	Heavy and viscid.
British thermal units per pound	19,291	19,086
Percentage of moisture in oil	trace.	trace.
silt in oil	under 1 per cent.	under 1 per cent.
Specific gravity at 60 degrees F.	.9322	.9322
Flash point, degrees F.	295	295
Burning point, degrees F.	295	295

The method of conducting tests was as follows:

The steam was brought to the desired pressure and the boiler was kept in use long enough to heat all parts thoroughly and to bring all conditions approximately to those under which it was desired to run the test. All observers were then called together and their watches were set to agree with the watch used by the Board. A time was set for starting the test, and the observers were sent to their stations and at the time set the level of water in the boiler, in the feed tank, and of oil in the feed barrel was noted. Data were observed at fifteen-minute intervals. A few minutes before the time set for ending the test all observers were notified when to take the final observation. The character of the smoke was observed by a member of the Board and is marked on a scale of 5, in which 5 denotes dense black and 1 a slight haze. Impeller boxes of burners not in use during any of the tests were blanked off by asbestos board. The boiler casing was kept as nearly tight as possible. It was necessary to replace one burner during the tests, and this was done in less than a minute. A small piece of waste was found in a groove in the washer of the defective burner.

Test No. 1 at 13.69 pounds of oil per hour per cubic foot of furnace volume. Capacity test.—Test begun at 11 A. M. and finished at 1 P. M., November 28, 1910. Weather, overcast. Steam jet in use during test. All burners in use. Rate of evaporation from and at 212 degrees F. per pound of oil: 13.70 pounds.

Test No. 2 at 7.85 pounds of oil per hour per cubic foot of furnace volume.—Test begun at 2:05 P. M. and finished at 5:05 P. M., November 28, 1910. Weather, overcast. Burners 2, 4, 5, 7, 8, 9, 10 and 11 in use. Rate of evaporation from and at 212 degrees F. per pound of oil: 14.37 pounds.

Test No. 3 at 5.54 pounds of oil per hour per cubic foot of furnace volume.—Test begun at 9:40 A. M. and finished at 12:40 P. M., November 29, 1910. Weather, overcast and raining. Burners 1, 6, 7 and 10 in use. Rate of evaporation from and at 212 degrees F. per pound of oil: 15.72 pounds.

Test No. 4 at 3.07 pounds of oil per hour per cubic foot of furnace volume.—Test begun at 1:25 P. M. and finished at 5:25 P. M., November 29, 1910. Weather, overcast and raining. Burners 1, 3 and 6 in use. Rate of evaporation from and at 212 degrees F. per pound of oil: 15.86 pounds.

The above tests were concluded on November 29, 1910, and the Board dissolved. Upon working up the data of the above tests roughly it was found that test No. 2 did not give the efficiency equal to that which would have been expected from points in a curve obtained from results of the other three tests. On this account two additional tests were made by the senior member of the Board, the civilian assistants of the Inspector of Machinery being present to take data. The results of these two tests are included in the report of the Board.

Test No. 5 at 8.86 pounds of oil per hour per cubic foot of furnace volume.—Test begun at 10:40 A. M. and finished at 1:40 P. M., November 30, 1910. Weather, clear. Burners 1, 2, 3, 4, 5, 6, 8 and 10 in use. Steam jet in use at intervals. Rate of evaporation from and at 212 degrees F. per pound of oil: 14.12 pounds.

Test No. 6 at 8.97 pounds of oil per hour per cubic foot of furnace volume.—Test begun at 10 A. M. and finished at 1 P. M., December 3, 1910. Weather, clear. Burners 1, 2, 3, 4, 5, 6, 8 and 10 in use. Steam jet slightly open during run. Rate of evaporation from and at 212 degrees F. per pound of oil: 15.44 pounds.

All conditions of the above tests were regulated by men in the employ of The Babcock & Wilcox Company who had been employed on this boiler for some time, making preliminary tests, and they were expert in handling this system of burning oil.

While the conditions existing were regulated by skilled men, the supervision of the Board was rigid, and the results obtained can be relied upon.

The Board has no knowledge of authenticated tests of fuel-

oil burning in which a boiler has been forced to the degree shown by Test No. 1, or in which an efficiency as high as that shown by Tests Nos. 3 and 4 has been attained. The results of the tests are, therefore, considered to be particularly impressive as indicating a material advance in the art of fuel-oil burning with mechanical atomizing burners. The variations in efficiency shown by Tests Nos. 5 and 6 indicate the careful adjustment of firing conditions that is necessary in order to obtain the highest efficiency when burning fuel oil.

The tests bring out the desirability of making gas analysis at frequent intervals when burning oil; also of watching the temperatures of the uptake closely. From the observation of the Board during these tests the character of the smoke was also an excellent guide to the results which were being obtained at any time. Changes in conditions were noted by the character of the smoke before they were apparent from the gas analysis. In Tests 3, 4 and 6 the character of the smoke was kept practically constant. It appears that the best results were obtained with the character of the smoke between 1 and 1½ on the scale used, and the corresponding percentage of CO_2 in uptake gases was about 12 per cent.

The Board was particularly impressed with the excellent results obtained with this boiler under the maximum rate of combustion, Test No. 1, which gives a combustion of 13.69 pounds of oil per cubic foot of furnace volume. This is the equivalent of about 75.34 pounds of coal per square foot of grate area in the same boiler when burning coal. The boiler in this test steamed freely with a very slight increase in the wetness of steam, and the falling off of efficiency was small for a rate of combustion much above the maximum ordinarily used on boilers of the Navy under forced-draft conditions.

After all the tests were completed the boiler was opened, cleaned and thoroughly inspected for deterioration. No tubes showed any signs of distortion, and all tubes and headers were free of blisters. All baffles were in good condition and properly placed.

OIL TESTS OF BABCOCK & WILCOX MARINE BOILER.

	1	2	3	4	5	6
Number of test						
Date of test	Nov. 28, '10	Nov. 28, '10	Nov. 29, '10	Nov. 29, '10	Nov. 30, '10	Dec. 3, '10
Duration of test, hours	2	3	3	4	3	3
Kind of oil			Texas	crude.		
Oil burner used	Pea	body	mech	anical	atomizers.	
State of weather	Over-cast.	Over-cast.	Rain.	Rain.	Clear.	Clear.
Barometer	...	...	...	...	...	...
Number of burners in use	11	8	4	3	8	8
Average Pressures.						
Steam pressure by gage, pounds	209.9	210.4	210.7	212	214.8	214.8
Oil pressure by gage, pounds	191.1	188.8	175.6	131.3	153.2	171.8
Draft pressure in fireroom, inches of water	2.60	1.69	1.18	.33	1.97	1.64
Draft pressure in furnace, Pass. 1, inches of water	...	...	...	.65	1.35	1.65
Draft pressure near uptake, inches of water	4.83	2.10	1.64	.72	2.58	2.79
Average Temperatures.						
Outside air, degrees F	45.5	45	43	46	43	...
Fireroom, degrees F	71.1	75.2	70.0	79.0	79 0	76.0
Steam (at gage pressure, tables), degrees F	391.5	391.7	391.8	392.3	393.4	393.4
Oil, degrees F	175.3	183.4	184.0	210.1	199.0	195.7
Feed water entering heater, degs. F.	47	47	47	47	46	46
Feed water entering boiler, degs. F.	168.6	160.9	201.0	211.2	185.6	182.8
Chimney gases, degrees F	771	666	533	447	702	630
Oil.						
Weight of oil used during trial, lbs.	5,943	5,112	3,605	2,665	5,767	5,840
Steam.						
Quality	99.189	99.290	99.837	99.891	99.782	99.835
Percentage of moisture	.811	.710	.163	.109	.218	.165
Smoke, scale of 5	2.1	1.5	1.3	1.5	2.3	1.15

OIL TESTS OF BABCOCK & WILCOX MARINE BOILER.

Number of test	1	2	3	4	5	6
Water.						
Total weight of water fed to boilers corrected for inequality of water level and steam pressure at beginning and end of test, pounds	74,898	67,036	53,464	40,096	75,714	83,573
Equivalent weight of water evaporated into dry steam, pounds	74,291	66,561	53,376	40,185	75,549	83,435
Factor of evaporation	1.096	1.104	1.062	1.052	1 078	1.081
Equivalent weight of water evaporated into dry steam from and at 212° F., lbs.	81,423	73,483	56,685	42,276	81,449	90,193
Oil Fuel per Hour.						
Oil per hour, pounds	2,972	1,704	1,202	666	1,922	1,947
Oil pr. hr. pr. cu. ft. furnace vol., lbs.	13.69	7.85	5.54	3.07	8.86	8.97
Oil pr. hr. pr. sq. ft. heating sur., lbs.	1.156	.663	.467	.259	.747	.757
Oil per hour per burner, pounds	270.2	213	300.5	222	240.3	243.4
Equiv. to coal pr. sq. ft. of G. S., lbs.	75.34	37.45	28.34	16.13	43.96	46 14
Water per Hour.						
Feed water per hour, pounds	37,449	22,345	17,821	10,024	25,238	27,858
Water per hour, corrected for quality of steam, pounds	37,146	22,187	17,792	10,046	25,183	27,812
Equivalent evaporation from and at 212° F. per hour, pounds	40,712	24,494	18,895	10,569	27,149	30.064
Equiv. evaporation from and at 212° F. per sq. ft. of heating surface, lbs.	15.83	9.53	7.35	4.11	10.56	11.69
Equiv. evaporation from and at 212° F. per cu. ft. of furnace volume, lbs.	187.60	112.87	87.06	48.70	125.10	138.53
Economic Results.						
Water evaporated per pound oil, lbs.	12.60	13.11	14.83	15.04	13.13	14.31
Equivalent evaporation from and at 212° F. per pound oil, pounds	13.70	14.37	15.72	15.86	14.12	15.44
Chimney Gas Analysis.						
Carbon dioxide (CO_2), per cent	9.85	9.26	11.57	11.86	10.71	10.94
Oxygen (O), per cent	6.46	7.68	4.50	4.08	5.18	4.73
Carbon monoxide (CO), per cent	.01	.00	.04	.04	.02	.00
Nitrogen (N), per cent	83.68	83.06	83.89	84.02	84.09	84.37
Efficiency.						
Efficiency of boiler	69.29	72.68	79.50	80.21	71.41	78.08

OIL FUEL.

By E. H. Peabody, Associate.

Ten years ago the discovery of the large oil fields of Texas, followed by the rapid development of the oil-bearing territories of California, marked an epoch in the history of the use of oil as fuel in America. Previous to that time the oil produced in this country had been of a character very valuable for refining purposes, so much so in fact, that the few oil-burning installations that sprang up with the development of each new field soon fell into a state of "innocuous desuetude" on account of a lack of fuel supply, and it was said that even the refiners themselves were burning coal because it was cheaper than oil.

In 1901, however, the Beaumont, Texas, oil wells, popularly called "gushers," on account of their enormous output, began flooding the market with a grade of oil not so valuable to the refiners as that previously produced and, therefore, cheaper for fuel purposes. This oil is heavier than the oils of Pennsylvania and the Middle West. It has a base of asphaltum, instead of paraffine, and carries a considerable quantity of sulphur; but is an oil ideal in every way for fuel purposes, barring the single possible exception that in its crude state it has a very low flash point and requires care in handling and storing. This objection has been overcome by removing from the crude oil some of its lighter constituents. Beaumont oil promptly practically replaced all other fuels in Texas, Louisiana and contiguous territories, and New Orleans became headquarters for the exploitation of every type of oil-burning device that had sprung up, flourished and finally become defunct in other fields.

Contemporaneously the oil lands of California, particularly the Kern River territory, began yielding ever-increasing quantities of petroleum, the quality of which was similar to the Texas product, though much of it was heavier, with slightly

less heat value. This oil was practically all available for fuel purposes and very rapidly the importation of coal in California became an obsolete industry.

ANALYSES OF CRUDE OIL.

Kind of oil.	Carbon.	Hydrogen.	Sulphur.	Oxygen and nitrogen.	Specific gravity.	B.T.U. per lb.
Pennsylvania.....	84.90	13.7	0.0	1.4	.886	19,224
Pennsylvania.....	82.0	14.8	0.0	3.2	.730	20,890
Ohio.....	85.46	14.2	0 34	0.0	...	21,209
Ohio.....	80.20	17.1	0.0	2.7	...	21,600
California	81.52	11.01	0.55	6.92	.966	18,667
Texas.....	84.6	10.9	1.63	2.87	.924	19,060
Texas.....	85.03	12.3	1.75	.92	...	19,100

Thus in the Southwest and on the Pacific Coast, and to a limited degree in other parts of the country, the burning of crude oil became a live proposition, and engineers and consumers began to inquire into methods and apparatus that would give the best results. All the burners then on the market atomized the oil by means of steam or compressed air and many of them were extremely wasteful and difficult to clean, their design and operation being very much of a mystery. Most of them had been developed in fields where light paraffine oils were produced and were consequently not adapted for use with heavy oils.

At this time the United States Navy appointed a board of engineer officers, headed by Capt. John R. Edwards, U. S. N., which made an extended series of tests with liquid fuel, the results of which have been quoted all over the world. The Babcock & Wilcox Co. also undertook an exhaustive series of experiments, under the writer's direction, covering a period of two years, first in Louisiana and later in California. Simultaneously many progressive engineers and manufacturers began using oil in stationary plants, in railway service and on board ship, and much valuable information became available.

It is characteristic and perhaps not altogether to our credit, that we in this country attacked the problem of burning oil in our own way and solved it to suit our own conditions, and

did it well, too, without paying much attention to what was being done abroad. Russian fuel oil had been on the foreign markets for years, and even in the early nineties methods of atomizing crude oil by centrifugal force were being developed there. We knew about this in the United States, and the idea of atomizing the oil without the use of steam or other medium at once appeals to the thoughtful engineer who has anything to do with oil burning. In spite of this, the Liquid Fuel Board only touched briefly on mechanical atomization and passed it up to the next generation. The writer confesses that after trying an improvised mechanical atomizer and finding that it worked about 200 per cent. better when he turned a little steam into it, decided for the time being to confine his investigation to burners of the usual steam or air-atomizing type. And this much must be said in justification: The simplicity of the steam atomizer and the excellent economy of the better types, together with the low oil pressure and temperature required, makes it a very strong favorite for stationary plants or, indeed, for any installation where the loss of fresh water is not a vital consideration. There are also many very efficient marine outfits in operation using compressed air for atomizing, and these are suitable for, and capable of, deep-sea cruising. However, for marine service, particularly naval service, the so-called mechanical atomizer has the preference. When the Bureau of Steam Engineering in 1907 included in one of its specifications the requirement for oil burners that they should be of the mechanical atomizer type, I consider that it marked the beginning of another epoch in oil burning in America. If not an epoch, certainly another chapter, and one in which is destined to be written the record of a distinct advance, not only in naval achievement but in the merchant marine as well.

We are on the point of seeing added to our Navy an oil-burning battleship. Last August the Engineer-in-Chief of the Navy, Rear Admiral Hutch I. Cone, said, before the United States Naval War College at Newport: * * * "Our experience with liquid fuel, while limited, has been sufficient to convince me that, considered from an engineering point of

view, the expediency of its use to the exclusion of coal, in battleships, is clearly indicated." * * * "I wish, gentlemen, to place myself emphatically on record as advocating oil-burning battleships for our service."

These are prophetic words. The advantages of oil fuel are manifest and have often been repeated. On given displacement or weight of fuel carried, oil, as compared with coal, increases the steaming radius about 50 per cent. On a given bunker capacity, *i. e.*, cubic feet of fuel storage, oil nearly doubles the steaming radius. Burners can be lighted and brought to full capacity in about two minutes. The fires never get dirty; there are no ashes; there should be little or no smoke; the labor of handling the boiler room "full speed ahead" is scarcely greater than when the ship is at anchor, and a hose over the sides will replenish the fuel supply or "oil ship," and the bunkers can be put in the double bottom. Also while we may be lacking in an adequate number of colliers for war service, the merchant marine is equipped with a fleet of oil-carrying tank ships which would all be available in war time for transporting oil fuel to the fleet.

The only argument against the oil-fuel proposition—the one great reason why the oil battleship is not already a reality—is lack of supply. Can we get the oil? In a recent report on the possibilities of the oil-bearing territory of this country, the United States Geological Survey published the following estimate, the probable ultimate output lying between maximum and minimum figures from the various fields.

GEOLOGICAL SURVEY ESTIMATE OF PROBABLE OIL PRODUCTION.

Fields.	Minimum. *barrels.*	Maximum. *barrels.*
Appalachian	2,000,000,000	5,000,000,000
Lima, Indiana	1,000,000,000	3,000,000,000
Illinois	350,000,000	1,000,000,000
Mid-Continent	400,000,000	1,000,000,000
Gulf	250,000,000	1,000,000,000
California	5,000,000,000	8,500,000,000
Minor fields	1,000,000,000	5,000,000,000
Total	10,000,000,000	24,500,000,000

This report estimates the aggregate oil-bearing area in twenty-two States as 8,450 square miles, 850 square miles of this being allotted to California. It does not include Alaska, where there are indications of large oil fields as yet undeveloped.

In a paper read at a meeting of the American Society of Mechanical Engineers last December, Mr. A. F. L. Bell, of San Francisco, shows that in only 700 square miles in California there will be a probable total production of nearly 17,600,000,000 barrels, of which less than 400,000,000 barrels have already been taken out. This is about double the maximum output estimated by the Geological Survey, which would, to say the least, indicate that the latter's figures of nearly twenty-five billion barrels for the United States, is conservative. All of this oil, it is true, is not available for fuel, though the proportion is large, and with such enormous figures before us we need have no fear for a fuel-oil supply sufficient to meet the needs of the Navy.

Through the courtesy of the Editor of the "Oil and Gas Journal," of St. Louis, I am able to quote some figures showing the state of oil production in the United States for the year 1910.

"California produced in 1910 at least 50,000,000 barrels of crude oil that is fit for use at the well, in gravity ranging from 11 to 25 degrees Baumé, much of it below 20 degrees, or not above that. Texas and Louisiana in 1910 produced at least 7,000,000 barrels of crude oil fit for straight fuel, gravity 17 to 23. Oklahoma and Kansas produced in 1910 about 2,000,000 barrels of crude oil fit for fuel, gravity 27 to 30 degrees. Illinois produced in 1910 about 1,000,000 barrels of crude oil fit for fuel, gravity 26 to 30. These figures are quoted in round numbers, to give an approximate idea of the straight fuel-oil production of the United States in 1910, with no expert or technical analysis of the quantity and varying qualities of the grades that fall within the degrees of gravity that permit use direct from the well.

"In residuum from refineries, both of strictly refining grades

and grades of straight fuel crude from which a small per cent. distillates have been extracted, the lump quantities may be set down about as follows: California, 10,000,000 barrels; Oklahoma, 15,000,000 barrels; Illinois, 8,000,000 barrels; Texas and Louisiana, 3,000,000 barrels; miscellaneous, 8,000,000 barrels; giving a total of 44,000,000 barrels of residuum for 1910, and a total of straight crude for 1910 of 60,000,000 barrels, the total production of all grades of crude in the United States in 1910 being 205,000,000 barrels."

The publication of the Bureau of Steam Engineering's specification calling on the shipbuilders to furnish oil burners of the mechanical atomizing type found American engineers unprepared. We still adhered to steam and air atomizers and they were doing well, but there was not an American burner on the market that would fulfill the conditions.

If there had been any disinclination before, to go abroad for advice, there was none now. A small coterie of distinguished engineering tourists at once took ship for England, and the result was that there are in the Navy today four different designs of mechanical atomizers and all of them foreign importations. Mostly they have done well, and credit is due to the men who have thus succeeded with a new thing here. But much has already been learned in our present brief experience about "what *not* to do," and errors which were of little moment in torpedo boat destroyers, or where oil was burned occasionally as an adjunct to coal, must not be perpetuated on an exclusively oil-burning battleship.

The conditions applicable to a high-powered cruiser or battleship of our Navy will not be found in any other navy in the world. We have some local color, and it is well that we have, for it is that alone which justifies our hope that our ships are, maybe, a little bit better than the other fellow's. Some study of the oil-burning problem as it applies to the projected "oil battleships" of our Navy is therefore justified, and it will be surprising if some good does not result therefrom.

Shortly after the appearance of those famous specifications of 1907 The Babcock & Wilcox Company commenced a series

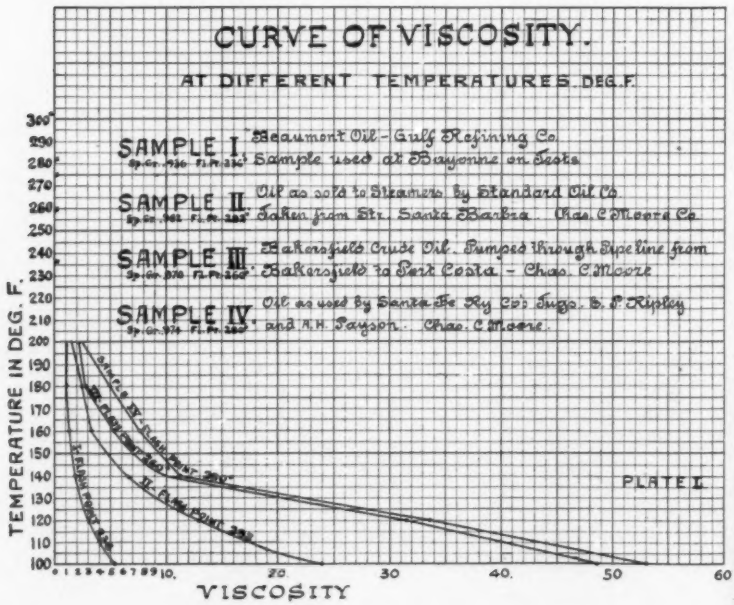
of experiments on burning oil with mechanical atomizers. Begun, first in connection with a brick "oven" furnace with no boiler, and continued under the B. & W. boiler of the steam yacht *Idalia,* they are still being carried on under a test boiler practically identical with the B. & W. boilers of the battleships *Wyoming* and *Arkansas.* Elsewhere in this issue of the JOURNAL will be found the results of tests made by a board of naval officers on this boiler, with oil fuel and with an American system of atomizers and furnace arrangement. These tests have been followed by experiments with coal and oil in combination, using both round and flat-flame oil burners. These have justified our belief that under service conditions it is extremely difficult, if not impossible, to get efficient results with either fuel if both are used in combination. Regardless of other things, oil improves the steaming capacity when burned with coal, but better capacity would be obtained by burning the same amounts of the two fuels in separate furnaces.

With mechanical atomization, as with steam atomization, the construction of the furnace and the method of admitting air is of paramount importance. The large furnace and sloping-roof baffle of the Babcock & Wilcox marine boiler have been found admirably suited to oil-fuel conditions, and very high capacities have been obtained with excellent economy and perfect distribution of the oil flame. It is a simple matter to prevent all smoke, but better results are obtained with a light cloud from the funnel, which would not, however, be visible at a distance of a few miles.

Contrary to a popular fallacy, the oil does not issue from the tip of the atomizer as a revolving spray, but the particles of oil fly off radially from the tip in the form of a cone. The air for combustion should preferably be given a whirling motion round the oil spray, the direction of rotation being practically immaterial, but the velocity and method of controlling and producing the whirling air current being of vital importance in its effect on the combustion, and also on the shape and character of the resulting flame. It should be the aim of the designer to obtain results with the lowest air pressure possible,

and the field for burning oil by this method with natural draft is already being invaded.

The oil should not be heated above its flash point. Besides being dangerous, high heating of the oil is not necessary. The action of the atomizer is distinctly mechanical, and evaporative tests have shown that the only advantage in heating the oil is in reducing its viscosity to a point where the centrifugal force imparted to the liquid will reduce it to a fine spray. The reduction in viscosity due to heating is very rapid at the lower


CURVE OF VISCOSITY.
AT DIFFERENT TEMPERATURES DEG. F.
SAMPLE I. Beaumont Oil - Gulf Refining Co.
Sp. Gr. .936 Fl. Pt. 315° Sample used at Bayonne on Tests
SAMPLE II. Oil as sold to Steamers by Standard Oil Co.
Sp. Gr. .901 Fl. Pt. 285° Taken from Str. Santa Barbra Chas. C. Moore & Co.
SAMPLE III. Bakersfield Crude Oil. Pumped through Pipe line from
Sp. Gr. .870 Fl. Pt. 250° Bakersfield to San Pedro - Chas. C. Moore
SAMPLE IV. Oil as used by Santa Fe Ry Co's Engs. D. F. Ripley
Sp. Gr. .914 Fl. Pt. 280° and A. H. Payson. Chas. C. Moore.
TEMPERATURE IN DEG. F.
VISCOSITY
PLATE I.

temperatures and some heating is necessary for the production of a spray of sufficient fineness. A temperature is soon reached, however, above which not only is there no improvement in the spray, but the capacity of the burner is greatly reduced. Plate I shows the effect of heating on the viscosity of the oil, and it will be observed that within 100 degrees of the flash point very little reduction in the viscosity takes place.

Heating the oil to a point within 50 degrees below its flash point is all that should ever be required.

The density of the oil is a decided factor, and atomizers which may handle very nicely indeed oil of 25 degrees Baumé may get into trouble with heavier oils such as 16 degrees or 18 degrees Baumé. The oil burner that goes into a battleship ought to be given a good dose of heavy oil before it is mustered into service.

DENSITY OF OIL.

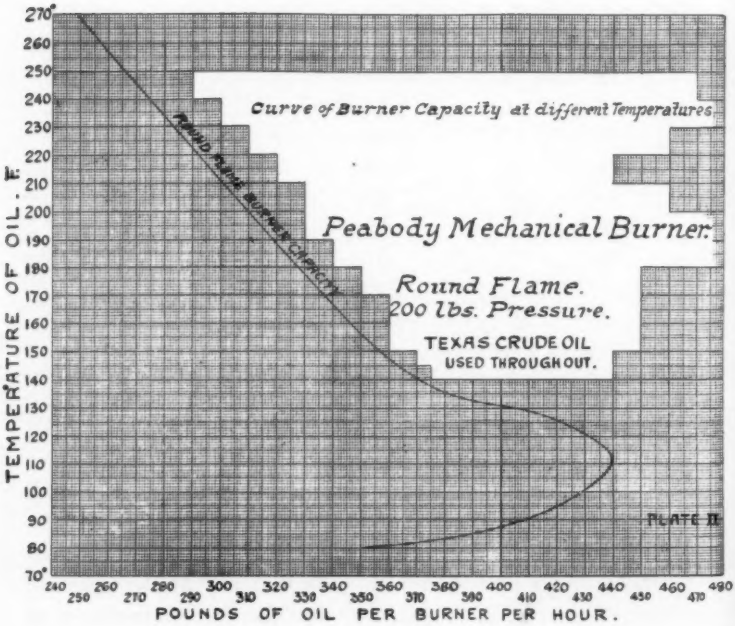
Degrees Baumé.	Specific gravity.	Pounds per gallon.
12	.986	8.22
14	.973	8.11
16	.960	8.00
18	.948	7.90
20	.936	7.80
22	.924	7.70
24	.913	7.61
26	.901	7.51
28	.890	7.42
30	.880	7.33
32	.869	7.24

With oil of low viscosity a good atomizer will work at 50 pounds pressure, but the spray will improve and become finer as the pressure is increased to 100 or 150 pounds. Increased pressure naturally increases the capacity of the burner.

As mentioned above, however, the capacity is also affected by the temperature of the oil. Assuming that the pressure remains constant, the capacity of the burner first increases as the temperature of the oil is raised to a point where the viscosity of the liquid is sufficiently reduced. After that continued heating of the oil results in lowering the capacity of the burner. This is shown clearly by the chart in Plate II.

The two factors, temperature and pressure, taken together, provide a wide range of capacity and form a ready means of controlling the steam output, in that they act alike on all burners in operation. Further, if the air or draft pressure is

also altered with the oil supply the desired steam pressure with the *maximum efficiency* is readily maintained. Any great increase or decrease in capacity should be taken care of by lighting up or extinguishing additional burners, each of which should have its individual air supply, so that it can be turned on full when the burner is lighted or shut off entirely when the burner is put out of service. Adjustments on the burner


Curve of Burner Capacity at different Temperatures.
Peabody Mechanical Burner.
Round Flame.
200 lbs. Pressure.
TEXAS CRUDE OIL
USED THROUGHOUT.
PLATE II.
TEMPERATURE OF OIL. F.
270° 260 250 240 230 220 210 200 190 180 170 160 150 140 130 120 110 100 90 80 70°
240 250 260 270 280 290 300 310 320 330 340 350 360 370 380 390 400 410 420 430 440 450 460 470 480
POUNDS OF OIL PER BURNER PER HOUR.

for regulating the quantity of oil are unnecessary and, in fact, objectionable. It requires a lot of labor to adjust the individual burners every time the steam pressure varies, and, furthermore, unless great judgment is displayed by the operator in adjusting the air supply at the same time, there will be either too much smoke and incomplete combustion, or excess of air entering the furnace, with a resulting loss in efficiency. A

slight adjustment at the oil pump or the heater, and the blower, will do the trick with economy and dispatch.

In conclusion, the writer wishes to point out that oil fuel, while presenting possibilities as to high capacities and high evaporative results quite incapable of accomplishment with coal, also offers to the careless operator opportunities for wastefulness which the worst coal-heaver cannot hope to approach. Quantities of oil can be burned under a boiler, to all appearances in an alluringly beautiful fashion, without making any steam at all. Under the same conditions a coal fire would go out.

Careful operation is a prime requisite for economical performance. Further, every point of boiler design which is conducive to efficiency with coal, applies equally to oil, and in addition, too much stress cannot be laid on proper furnace design and arrangements for admission of the air for combustion. Fortunately there is no mystery about it; simplicity of construction is still possible and preferable, nor do we need the higher mathematics or very much abstruse theory to prophecy success for the oil-burning battleships of the future.

FRICTION AND LUBRICATION.

By Lieutenant G. J. Meyers, U. S. N.,

School of Marine Engineering.

(Comment and suggestions for further investigation are invited.)

FRICTION.

When two surfaces are pressed firmly together they cannot be moved relatively to one another without the exercise of considerable force. When, by applying sufficient force, the surfaces have commenced sliding one over the other, the resistance to be overcome in order to keep them in relative motion has at all speeds a considerable retarding effect. This resistance is called the friction between the two surfaces. Friction, then, is that resisting force which always acts to prevent and to retard the relative motion of one particle or body in forced contact with another particle or body.

Friction acts at the surfaces of contact of the two particles or bodies between which it is exhibited, and in the direction of their common tangent, resisting relative motion in whatever direction it may be attempted to produce motion. Friction is thus always a resisting force.

Friction is of two kinds: (1) Solid friction, that exhibited between solids; and (2) fluid friction, that due to the movement of particles of liquids or gases over each other or over solid bodies. Solid friction is of two kinds: (1) Sliding friction, where one solid slides over another; and (2) rolling friction, where one solid rolls over another.

I.—*Sliding Friction.*

The friction between solids is caused by the roughness and unevenness of the surfaces in contact. In the case of sliding friction, the asperities of one surface interlock with those of

the other, and motion can only take place by the sliding of the one set over the other, by the tearing off of the projecting parts, or by rubbing them down. In either case the process gives rise to a resistance, which is the greater as the roughness is greater and material softer, and the less as the surfaces are smoother and material harder, in each case depending on the amount of foreign substance interposed between the surfaces. This resistance increases as the pressure holding the two surfaces together increases, and reaches a maximum when the two surfaces due to this pressure become welded together. In this case the maximum resistance becomes the shearing strength of the metal at the point of welding.

Sliding friction may be measured, and as measured is treated as a force retarding motion. The relation borne by this force to the load or force pressing the surfaces together has been the subject of much speculation and discussion, and although numerous experimental researches have been carried out, there has always been a conflict of opinion. The generally accepted relation, called "the coefficient of friction," is the ratio between this retarding force, or the force necessary to move an object or keep it in motion, and the force pressing the surfaces together. Thus if a weight of 2,240 pounds of one substance be placed on a horizontal plane of another substance, and a force of 350 pounds be required to move it, then the coefficient of friction is $\frac{350}{2,240} = 0.156$, or, as commonly expressed, the relation $\frac{F}{W} = u$, where F is the resistance due to friction, or is the force applied necessary to produce motion, W is the force pressing the surfaces together, and u is the coefficient of friction or the friction per unit of load.

The force required to start a body from rest along a horizontal plane surface differs somewhat from the force required to keep the same body in motion over the same plane. This will give varying values for the coefficient of friction. The first is called the "static coefficient of friction," and the second the "kinetic coefficient of friction."

The power which is expended in overcoming the frictional resistance of two rubbing surfaces and in keeping them in motion is converted into heat, and increases the temperature of the masses in contact. When the amount of friction is small, the heating of the masses is also small, but should the frictional resistance become large from any cause, the heating of the materials in contact may become so great that the opposing surfaces become brittle, melt, or even weld together.

It is clear that some relationship exists between the work done and the heat produced. In the case of sliding surfaces in contact under pressure,

W=Total pressure in pounds on the rubbing surface;

u=Coefficient of friction;

V=Velocity of rubbing in feet per second:

R=Force of friction;

then

$$\text{work done}=WuV=RV, \text{ where } u=\frac{R}{W},$$

and

$$\text{B.T.U.'s of heat developed per sec.}=\frac{WuV}{779}=\frac{RV}{779}.$$

The heat generated then by two bearing surfaces in relative motion is proportional to the friction overcome. The actual rise of temperature of the bearing surfaces depends not only on the rate at which heat is generated, but also upon the conductivity of the metals in contact and radiation from other exposed parts, so that frictional resistance cannot be proportional to the rise in temperature.

The maximum value of the force of friction in practice is that value where seizing begins. Values below this point depend on the pressure holding the surfaces together and consequently on the amount of abrasion, the temperature of the masses, the smoothness of the surfaces, the amount and character of foreign substances interposed between the surfaces, and the structure of the two surfaces. This probably accounts for the different values of the coefficient of friction for the same substances found by different experimenters:

the surfaces had different degrees of smoothness and cleanliness, temperatures were not kept constant, and the substances differed somewhat in structure.

The kinetic coefficient of friction for different substances has been found by experiment with machines of various designs, usually with a disc of one material attached to a vertical shaft and moving over a surface of the same or different material, the force of friction being measured by a lever with a movable weight attached to the lower surface with the weight adjusted at a known point just sufficient to balance the force of friction and keep the lower surface from revolving. The weight pressing the disc on to the surface may be varied by placing weights on the upper end of the shaft. Or the friction may be measured by a bearing placed on top of a

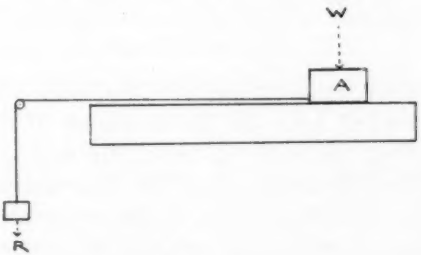


Fig. 1.

revolving horizontal shaft and the force of friction measured by moving a weight along a graduated beam attached to the bearing, to such a point that the bearing will not revolve with the shaft. The force pressing the surfaces together may be varied by placing different weights on the top of the bearing. In all cases it has been found that the coefficient of friction will vary for pressure, temperature, cleanliness and smoothness or fit of the two bearing surfaces and the structure of the two substances in contact.

Frictional resistance is approximately proportional to the load, is almost independent of the area of surface, and, except at very low speeds, decreases as velocity increases.

"The coefficient of friction is not constant for varying

speeds, but decreases materially with velocity, is very much greater at minute velocities of 0+, falls very rapidly with minute increases of such velocities, and continues to fall much less rapidly with higher velocities up to a certain varying point. It is practically constant for varying loads and is independent of area of the surfaces in contact provided seizing does not take place."

The most common method of measuring the coefficient of static friction is by starting from rest one surface placed in contact with a stationary horizontal surface as indicated in Fig. 1. The load is varied by varying the weight on the upper or movable surface (A). In this, weight (R) is the static friction if it just starts (A) from rest, and the coefficient of static friction is $\frac{R}{W}$, neglecting friction of pulley and cord.

II.—*Rolling Friction.*

Rolling friction, or resistance to rolling, is a consequence of the irregularities of form and the roughness of the surfaces of bodies rolling, one over the other. Its laws are not definitely established, in consequence of the uncertainty that exists in experiment as to how much of this resistance is due to (1) roughness of surface; (2) original and permanent irregularity of form; (3) distortion under the load; and (4) sliding friction. It is customary to take the resistance as proportional directly to the load and inversely as the radius of the rolling cylinder or wheel, or

$$R=\frac{W}{r}u,$$

where R=frictional resistance,
u=coefficient of friction,
W=load, and
r=radius.

Then

$$u=\frac{Rr}{W},$$

where the effort applied is at the axis as in the case of a wagon wheel.

When the effort acts at the circumference of the roller or wheel, as where straight-lined surfaces have relative motion on interposed rollers, the force of friction becomes

$$R=\frac{\frac{1}{2}M\,W_1}{r} \text{ and } M=\frac{2Rr}{W}.$$

The friction is unaffected by change of temperature.

The coefficient of friction of ball bearings at high speeds is rather higher than that of ordinary bearings well lubricated, but is less than that of ordinary bearings imperfectly lubricated. The friction of ball and roller bearings is small at very low speeds and with small loads, much smaller than ordinary bearings perfectly lubricated.

In all cases where ball bearings are used, it is absolutely imperative that there should be freedom from shock, for a smart blow will either break the balls or destroy the accuracy of the surfaces upon which they run.

The expression for friction of ball and roller bearings given above is subject to wide variations due to the different angles of the race-way surfaces to the instantaneous axis of revolution (see Fig. 2). As a rule, the smaller these angles the less

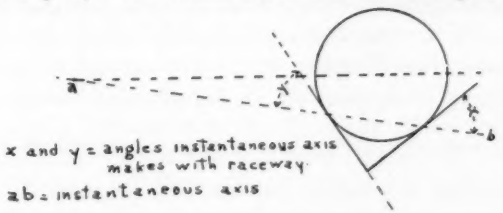


Fig. 2.

is the friction, as in this case the sliding friction is reduced.

Just what difference there is between the friction of ordinary bearings and ball bearings under similar conditions seems to have been subject to very little experiment, and will vary widely with design and thoroughness of lubrication.

III.—*Fluid Friction.*

Fluid friction is due to the movement of liquids or gases over each other or over solid bodies. When a fluid moves in

contact with a solid or when it flows in a current through a mass of fluid, precisely the same conditions arise. In either case the resistance experienced is due to the relative motions of layers of fluid moving in contact with each other, provided the solid surface is wetted with the liquid.

Much of what is written on fluid friction is speculation, most writers confounding viscosity as determined by the viscosimeter with fluid friction. So far no relation has been found to exist between the two. There seems to be some agreement, though, on the fact that though viscosity does enter somehow into the determination of the frictional value of a liquid, there are other properties that have a greater influence on a liquid's frictional value. These are greasiness, oiliness, unctuousness, or "body," and surface tension. Experiments, however, to determine what these values are and their effects on liquid friction have not been conducted.

There is no question, though, that the friction of liquids is much less than that of solids; this being proved by interposing fluids between solid rubbing surfaces and thereby reducing the friction.

Opinions differ as to the laws of fluid friction, but there seems to be no doubt that friction of fluids (1) decreases with rise of temperature, (2) is proportional to the area of the surfaces in contact, (3) is independent of the pressure or load, and (4) follows a condition expressed by $R=\frac{aVn}{s}$, when flowing in narrow and confined channels such as capillary tubes below a certain velocity called the critical velocity; and a condition expressed by $R=\frac{nDAV^2}{2g}$ when flowing in large tubes and above the critical velocity.

Thurston says in "Friction and Lubrication": "Fluid friction * * * is probably independent of pressure. In hundreds of experiments with pressures varying from 5 to 275 pounds to the square inch, I have found friction of well lubricated journals to vary in such a manner as to prove that the law which governs such cases is quite different from that

which applies with dry solids, and have thus found the deduction above given confirmed." And further in "Friction and Lost Work," Thurston says: "The friction of any fluid is found to be independent of the pressure, as first shown by Coulomb * * * and by many later observers * * *" In discussing "A New Oil-Testing Machine and Some of its Results," by A. Kingsbury, Thurston says: "It follows that, since the product of the coefficient of friction and the pressure is, under such circumstances, practically constant, the total frictional resistance, under similarly perfect conditions, is unaffected by change of load on the journal." Unwin in "Elements of Machine Design," part I, taking Tower's results as a basis, arrives at the expression that total friction divided by area, $\frac{R}{dl} = c_1 \sqrt{V}$, an expression that does not seem to be exact for velocity, but which concludes that friction of fluids is independent of load.

That fluid friction as applied to lubrication varies as the velocity seems to be well established, though the statements of various writers show a disagreement as to the acceptance of this law. Archbutt and Deeley, in "Lubrication and Lubricants," state that "carefully conducted experiments by Coulomb, Poseuille and others, on the rate of flow through capillary tubes, have proved conclusively that the resistance under certain conditions is proportional to velocity," and "the critical velocity is rarely if ever reached by lubricating films." Watson, in "Text Book on Physics," states that R, the tangential resistance to motion of a layer of liquid moving over another layer of liquid, is proportional to, $R = \frac{Van}{s}$,

where V=velocity,
a=area of surface,
s=distance of surface from fixed plane,
n=coefficient of viscosity,

where n is found from Poiseuille's formula,

Volume of liquid passed per sec.$=\frac{\pi pr^4}{8ln}$.

O. Reynolds (*) states "in the definite experiments made by Coulomb, and particularly by Poseuille, it was found that resistance (in liquids) was proportional to velocity."

Confusion on this subject can be avoided by considering on what data deductions are based. Most authors attempt to deduce laws of fluid friction from results obtained in experiments with lubrication where the conditions are not normal. In particular the velocity is different and the friction of the fluid is not measured by the movement of one layer of fluid over another, with all particles moving in parallel directions. Some of these erroneous deductions are based (1) on imperfect lubrication, where the supply of lubricant is so limited that the metals touch and the resulting solid friction confuses the data obtained; (2) failure to consider the fact that the particles of liquid are not confined at the sides and that they move longitudinally as well as transversely; (3) failure to consider the fact that the temperature of the entering fluid as well as the temperatures of the bearing, journal and interposed liquid film affect the results.

LUBRICATION.

The most serious loss of power in transmitting energy by mechanism of any kind is that due to frictional resistance between rubbing surfaces. Peabody states that the loss of power due to friction in a steam engine varies from 5 to 15 per cent. of its I.H.P.

It has been found that the frictional resistance of two solid surfaces rubbing on each other will be reduced by interposing between them certain foreign substances. Thus if the surfaces are contaminated by dirt, a film of oxidized metal, or a very slight film of grease, they will have less friction than if they are perfectly clean. But where the foreign substance interposed is solid, it is soon rubbed away and a new supply must be provided. But two rubbing surfaces will not carry

*"Theory of Lubrication," by Prof. O. Reynolds, Philosophical Trans. of the Royal Society, Vol. 177.

solid substances in between them by the mechanical action due to the motion of one or both. Where a liquid is interposed between the rubbing surfaces the friction will be reduced to approximately that of the liquid so long as the two surfaces do not touch. But the liquid is soon squeezed out from between the surfaces, due to the load they carry, and a new supply must be applied. The case of a new supply of liquid differs from the case of a new supply of solid in that, with the liquid, the mechanical action due to the motion of one or both surfaces will cause to be continually maintained between the rubbing surfaces a film of liquid if the liquid is applied in sufficient abundance to one or both of the moving surfaces.

The theory of lubrication may then be stated to be the interposition between two rubbing surfaces of some suitable liquid, the supply being constantly renewed by the mechanical action of the moving surface, so that a continuous film of liquid is maintained between the surfaces, reducing the friction from that of solid rubbing on solid to that of a layer of liquid rubbing on a layer of liquid, or to the resultant of solid rubbing on solid in spots and of liquid rubbing on liquid in other spots, depending on the perfection or imperfection of the supply of liquid and its quality.

Air has been successfully used as a lubricant by Prof. A. Kingsbury,* the coefficient of friction being found to be 0.00075, where the load was the weight of a journal revolving on half of a bearing. When the journal was started, the metals rubbed on each other and it required considerable force to maintain motion, but as the speed increased, a film of air formed between the journal and brass, and the force required to maintain speed and consequently the frictional resistance diminished until the journal was slowed down, when at a certain point the film broke down.

In practice the limits of friction for a bearing are a maximum at that point where the softer of the two metals starts to seize, and a minimum at that point where the two surfaces are entirely separated by a film of lubricant of such quality that

*Chemical Trade Journal, Vol. XXVI, p. 232.

this film will not break down under the heaviest loads that will be applied to that particular bearing. All results between the maximum and minimum are due to imperfect lubrication; that is, the amount of friction is greater or less depending on the solid surfaces touching to a greater or less extent and consequently producing a greater or less amount of abrasion.

In order, then, to reduce the friction of rubbing surfaces to a minimum, the ideal to be attained is that condition where the

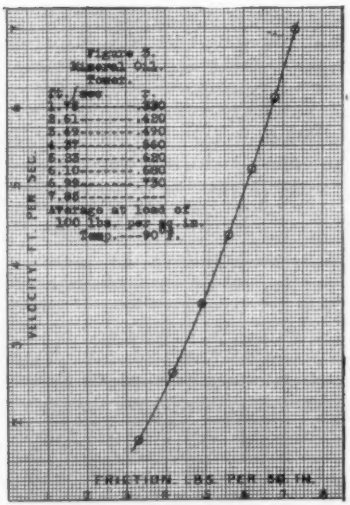


Fig. 3.

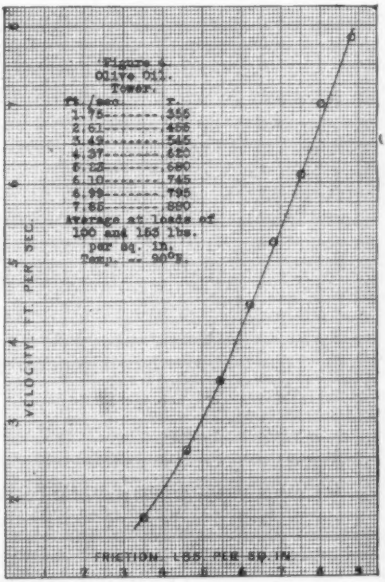


Fig. 4.

film of lubricant is constantly maintained between the surfaces, this film constantly keeping the two surfaces apart and preventing them from rubbing one on the other.

There is more difference of opinion concerning the laws of lubricated surfaces than on the laws of fluid friction, but those laws which have the best evidence to support them are as follows:

(1) Friction of lubricated surfaces decreases with the rise

of temperature of the bearing and journal, of the film of interposed lubricant, and of the supply of lubricant.

(2) Friction of perfectly lubricated surfaces is proportional to the area of the rubbing surfaces between which the film of lubricant is interposed.

(3) Friction between perfectly lubricated surfaces is inde-

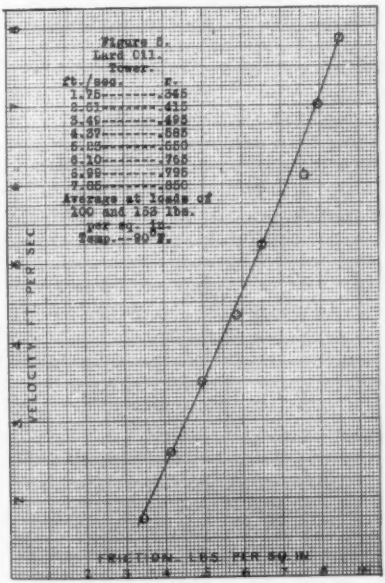


Fig. 5.

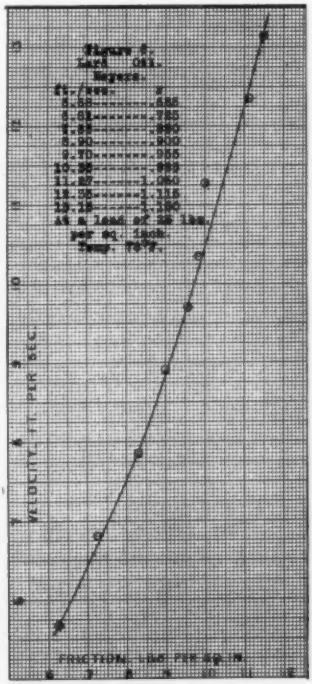


Fig. 6.

pendent of the load up to where the film starts to break down and the surfaces begin to touch.

(4) Friction between perfectly lubricated surfaces bears the following relation to speed:

$$r = \frac{R}{A} = \left(K + K_1 V^n\right).$$

where A=area of surfaces between which film is interposed,

R=friction of lubricated surfaces,

r=friction in pounds per square inch,

V=speed in foot seconds,

K and K_1 are constants depending on quality of lubricant, and its temperature,

n is a constant found to be approximately .8 from the results of Tower's experiments for mineral grease, mineral oil, lard oil, olive oil, for speeds from 2 feet to 7 feet per second, and for lard oil from results by the author from 7 feet to 13 feet per second.

The equations for the above-mentioned oils are as follows:

Tower .	Mineral grease	$r=.1596+.2154V^{.8}$	(1)
	Mineral oil	$r=.1626+.1198V^{.8}$	(2)
	Lard oil	$r=.10688+.14806V^{.8}$	(3)
	Olive oil	$r=.1185+.1491V^{.8}$	(4)
Meyers .	Lard Oil	$r=.0845+.1402V^{.8}$	(5)

Substituting values of V from zero to 7 in the first four equations, the results are plotted in figures 7, 8, 9 and 10. Substituting values of V from zero to 13 feet per second in the fifth equation, the result is plotted in figure 11. In all the curves both calculated and actual results are plotted. The close coincidence of the actual results with those computed shows that the relation $r=K+K_1V^{.8}$ holds very closely, and shows that the ideal lubricant would have $K_1=0$, with constant temperature.

The references previously given to Unwin and Thurston as to relation between frictional resistance and load or pressure in regard to fluid friction, apply as well in the case of perfectly lubricated bearings (see p. 7).

The relation between friction and speed in perfectly lubricated bearings has been in some doubt. The relation given above is deduced from Tower's experiments plotted in Figures

3, 4, 5, and from experiments made by the author on a Riehlé testing machine, the results for lard oil being plotted in Figure 6.

It would appear from this relation, $r=K+K_1V^n$, that when V is zero, $r=K$, and that consequently K is the static friction. This would indicate that the static friction is less than the kinetic friction. That this is possible and probable there seems to be no doubt. Archbutt and Deeley state:

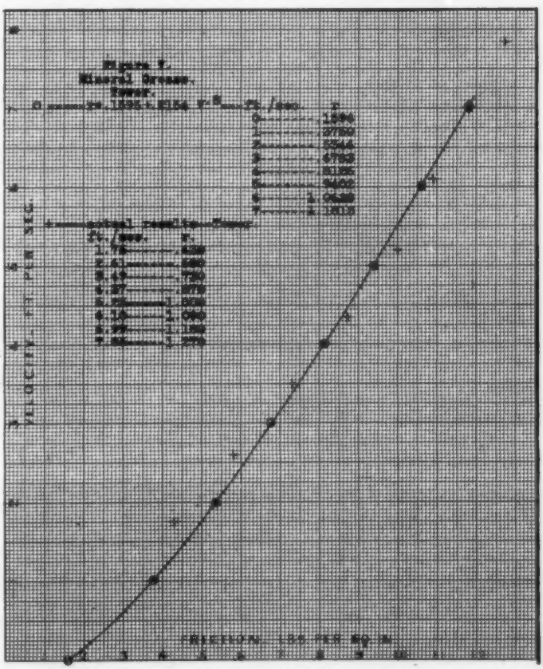


Fig. 7.

"From a number of experiments made by Kimbal, it would also appear that although there is no abrupt transition between the static and kinetic coefficients, the friction of motion is not always less than the friction of rest. This is more particularly the case when the loads are small or the surfaces are lubricated."

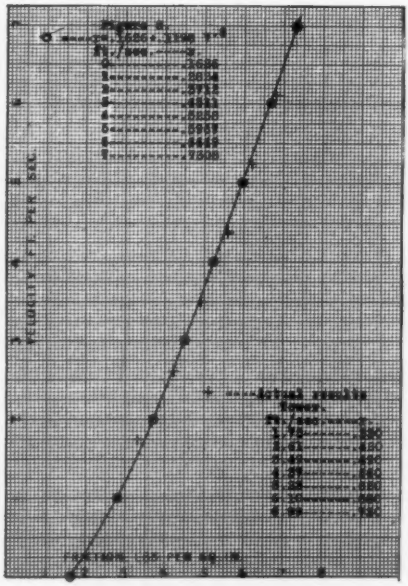


Fig. 8.

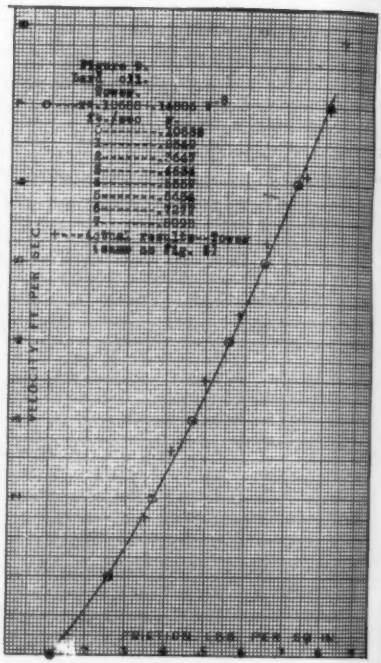


Fig. 9.

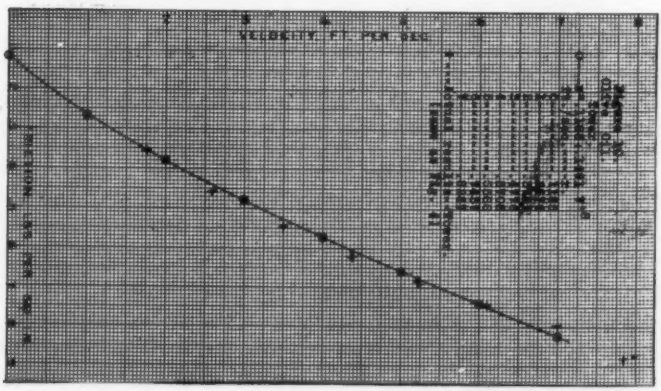


Fig. 10.

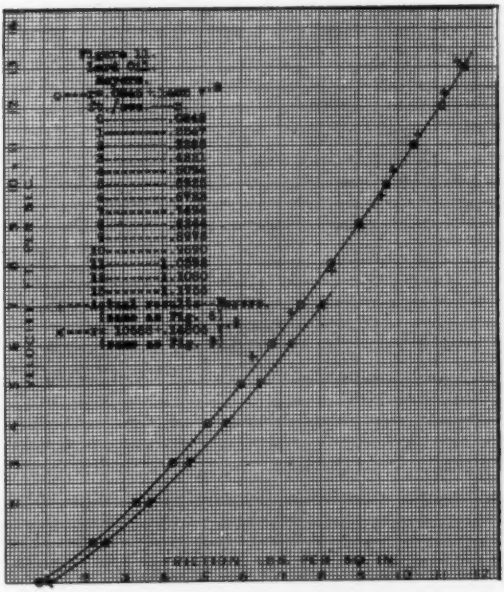


Fig. 11.

I.—*The Action of a Lubricated Bearing.*

B. Tower experimented on actual bearings lubricated with an oil bath and obtained very uniform results. Since his time numerous experiments have been conducted on lubrication, but have been lacking in value for the reason that the supply of oil was always limited, the results giving fluid friction combined with solid friction and showing no regularity when repeated under the same conditions, excepting tests made by A. Kingsbury in 1903. Osborne Reynolds worked out a mathematical theory of lubrication which when applied to Tower's results agreed very closely.

One important fact discovered by Tower was the distribution of pressure on a perfectly lubricated bearing. This is illustrated in Figure 12, where the bearing was placed on the top of the revolving journal and the load was made to

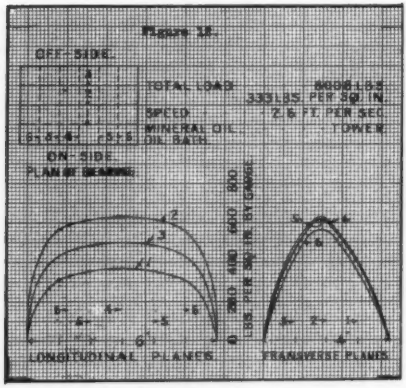


Fig. 12.

act in a vertical line through the center of the bearing. This condition would not be obtained in a reciprocating engine where the load is alternating first on one side then the other, except perhaps momentarily, but would be obtained on a turbine bearing. Figure 13 shows a curve of loads and pres-

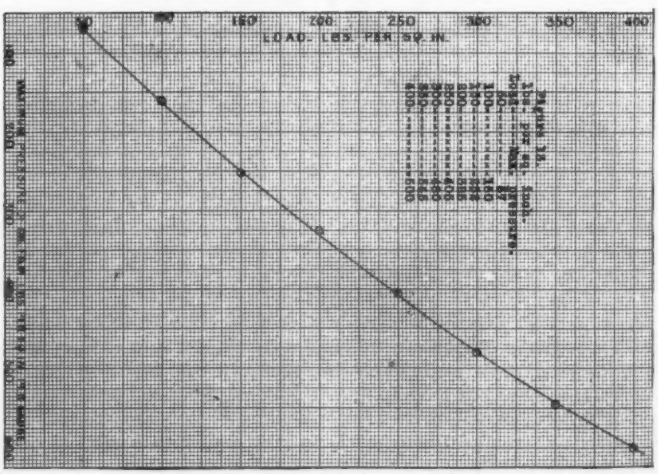


Fig. 13.

sures obtained by attaching a pressure gage to the center of the bearing of a Riehlé machine. It will be noted in both Figs. 12 and 13 that the maximum pressure recorded far exceeds the load per square inch on the bearing. It also shows the folly of attempting to introduce oil at any point of the bearing without considering in what direction the load is acting or what is the character of the action of the load.*

Referring to Fig. 12, it is seen that the pressures are zero at the ends where the oil runs out of the bearing, and zero also at the on-side and at the off-side edges. The resulting

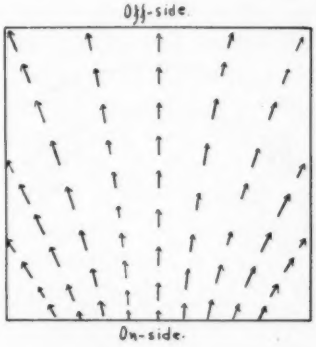


Fig. 14.

pressures are due to forcing the oil between the bearing and journal by no other means than the mechanical action of the revolving journal. This action caused the film of oil to form and be maintained between the bearing and journal so long as the supply of oil was abundant, and the bearing was allowed to take all the oil it could use.

From the fact that oil enters on the on-side and is carried

* Professor John Goodman states in "The Friction and Lubrication of Cylindrical Journals," "Subsequent experiments of the author on other machines entirely confirmed the results of the original experiment, and most clearly demonstrated that the practice of putting oil holes or grooves in brasses along the resultant line of pressure is radically wrong, and should be studiously avoided," and also states in the case of "a brass where an oil groove is cut along its crown, and an oil hole is drilled through the top of the brass into it." "This method of cutting oil grooves cannot be too strongly condemned; for not only is the friction enormously increased, but a bearing cannot be lubricated in this fashion if a syphon only is used."

in between the rubbing surfaces by the motion of the journal, and is ejected at the ends of the bearing, and is carried out to a certain extent on the off-side by the motion of the journal, the particles of oil take paths somewhat like those shown in Fig. 14, from the on-side to the ends and off-side of the bearing. It will thus be seen that the thickness of the oil film at the off-side will be less than on the on-side, the length of the journal being the same at all points. The journal will then approach closer to the brass at some point H, Fig. 15, on the off-side depending on the load on the bearing.

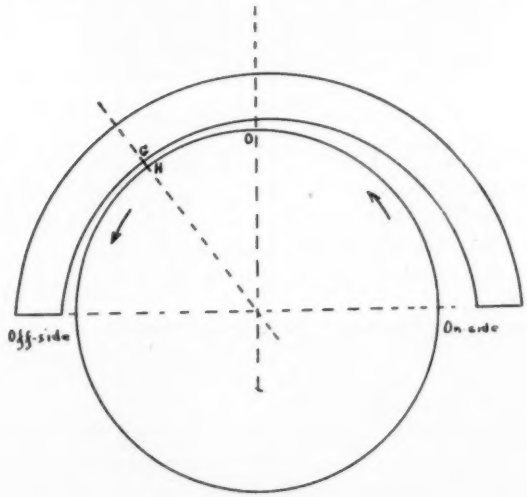


Fig. 15.

In the case of a reciprocating journal, such as a crank pin, the conditions as given above will only momentarily be attained as the load comes on the top or lower brass. If the oil is supplied in large enough quantities, then when the pressure or load is relieved on one brass, the oil should immediately fill the clearance space between the brass and the crank pin and the film should have formed when the load again comes on that brass. By also taking advantage of the movement of

the crank pin to mechanically form the oil film, additional assurance is had that the film will be formed by the time the load again comes on the brass.

II.—*Wear and Heating of Journals and Brasses.*

Before the journal starts to rotate the effect of the load will have brought the brass into contact with the journal at 0, Fig. 15. At starting the surfaces will be in contact, and the initial friction will be between solid surfaces, causing some abrasion. If oil is not immediately applied the friction will continue between solid surfaces, causing continued abrasion and heating. After motion commences the surfaces gradually separate as the velocity increases, and if the supply of oil is abundant the oil film gradually forms. But the oil film does not immediately form even with an abundant supply; and between the time of starting and until the oil film is established the friction is between solid surfaces, and there is some wear of the brasses. If, on the other hand, the supply of oil is limited when starting, the oil film will sometimes take hours to form and the time that the friction is between solid surfaces is increased and wear is greatly increased. This was definitely demonstrated by the author on a Riehlé testing machine. The oil supply was limited and the indications of the oil film forming were shown by pressure gage, as previously described. In one case, as an example, it took three hours and fifty minutes for the oil film to form, during which time it seemed to form and break intermittently, the pressure starting up slowly then suddenly dropping to zero. In the meantime, by varying the load, the friction increased approximately with the load, showing that it was solid friction. When starting machinery, then, every bearing should be furnished with an abundant supply of oil, for if the oil supply is sufficient, the time it takes to establish the oil film is reduced and consequently solid friction and wear are reduced. The time taken to establish the oil film with various loads, starting the journal from rest with an abundant supply of oil on the

journal at starting, varied from 4 minutes 30 seconds to 7 minutes 30 seconds on the Riehlé machine used by the author.

If, after starting, the supply of oil is limited; that is, the journal is not getting sufficient oil to establish an oil film, friction is going to continue between solid surfaces to a greater or less extent depending on the supply of oil, and abrasion will continue as long as the machinery is running and the supply of oil is insufficient. And although the supply of oil is insufficient for the journal to maintain an oil film, the temperature of the bearing may not become excessive, unless there are abnormally bad conditions of fit between journal and bearing. This case is well illustrated where an engine is lubricated by wick feed, and the number of strands of worsted is carefully worked out for each speed on the basis of keeping the temperature within working limits. Great economies of oil are attained, but it is not taken into consideration that the bearings are wearing excessively and that the cost of the oil saved will not pay for new bearings nor for the cost of having to lay up the machinery for repairs.*

Professor Goodman states: "Many people imagine that as long as a bearing occasionally gets a few drops of oil from an oil can that it ought to run cool and with but little friction, but this is a most fallacious idea, both from a frictional and economical point of view," and he further states: "Syphon lubrication and all other intermittent systems of dropping on to bearings are most barbarous, they not only waste an enormous quantity of oil but are very inefficient methods of reducing friction. The intermittent action causes the friction to vary most widely, to such an extent that it is quite out of the question to attempt to get reasonably regular results with this system."

*In the case of a ship in the Navy in 1906, that was held up as a model for oil consumption on the main engines, the number of wicks for each speed was carefully adjusted and the ship was a great oil saver. But in a four-hour, full-speed run the same ship attempted to save oil by the same method, and melted the white metal from one main bearing and had other hot bearings on both main engines, that necessitated slowing down and limping into port with one engine. No record of wear of bearings was kept, but the above illustration shows of what use this ship would have been in action under similar conditions.

If, on the other hand, with an abundant supply of oil, the condition of steady lubrication has been attained, the surfaces are completely separated by oil, and there should be no wear. But if there is wear, as there generally is from one cause or another, it would take place most rapidly where the surfaces are nearest, that is, at GH on the off-side of O (Fig. 15).

Thus while the motion is in one direction, the tendency to wear the surfaces to a fit would be confined to the off-side of O, along GH (Fig. 15). As the position of GH varies with the load, the wear would be distributed over an arc of small extent on the off-side of O, if the load were varied. This appears to offer a very simple and well-formed explanation of the important and common circumstance that new surfaces do not behave so well as old ones; and of the circumstance observed by Tower, that in the case of the oil bath, running the journal in one direction does not prepare the brass for carrying a load when the journal is run in the opposite direction.* This does not, however, take into consideration the effect of a misfit in the journal and brass.

If the surfaces are not strictly parallel in the directions TU and VW (Fig. 16), the pressure would be greatest in the

Fig. 16.

narrowest parts, causing axial flow from those into the broader spaces. Hence, if the surfaces were considerably irregular, the lubricant would, by escaping into broader spaces, allow the brass to approach and eventually to touch the jour-

* In the case of the U. S. S. *Virginia*, where the engines were changed from in-turning to out-turning, "the bearings were not changed, and gave trouble from the first * * *. After the full-power trial the main bearings and crank-pin brasses ran so hot that renewal was a necessity." This was due to the fact that the bearings had worn to a good working fit in the ahead direction for in-turning, and had not worn to a good fit for the backing direction for in-turning; in other words, the ahead direction for out-turning.

nal at the narrowest spaces, and this would be particularly the case near the ends where the pressure is zero.

It may be assumed generally that inequalities exist in the surfaces of new brasses and journals. And as the surfaces, according to their material and manner of support, yield to pressure, the brass will close on the journal at the ends where, owing to the escape of oil, there is no pressure to keep them separate. The section of new brass and journal lubricated will therefore be as shown in Fig. 16, the thickness of the film instead of being uniform will vary from zero to the maximum, being less at the ends. In this condition wear will take place at the points of contact, so that if the journal runs in one direction only the wear will be at the ends and in the vicinity of GH (Fig. 15), during which wear the friction will be great and attended with more or less heating and possibly serious abrasion. So long, however, as the journal runs in one direction only, the wear will take place on the off-side (at GH, Fig. 15), and the bearing will be worn to fit. In the meantime the bearing on the on-side will not be similarly worn, so that if the journal is reversed, the wear on the on-side will begin and the bearing will again heat more or less. Even in the case of a perfectly smooth bearing the running of the journal under a sufficient load in one direction should, supposing some wear, according to this theory, render the bearing less well able to carry the load when running in the opposite direction.

Taking the same conditions of perfect lubrication and supposing the oil film to be formed and maintained, in actually fitting a brass on a journal, it was found by the author that a pressure gage attached to the brass indicated that the oil film had formed and was being maintained while the journal was running, but that the friction was more than it should have been, due to brass rubbing in spots on the journal as the result of poor fitting. The projections on the brass were probably so small as to be immeasurable, yet pierced the oil film and caused excessive friction, heating and abrasion or wear. That this condition exists will be seen when it is

considered that Reynolds measured the thickness of the oil film and found it to be about 0.000375 of an inch, the measurements of Kingsbury giving 0.00021 to 0.00023 of an inch, both in the case of perfectly lubricated bearings.

Summing up the causes of heating and wear of bearings and journals, they may be stated as follows:

1. Poor fit of bearing on journal;
2. New bearings, whether properly fitted or not;
3. Not running machinery in both directions when new to insure a good working fit for both directions of running when machinery is working under ordinary conditions;
4. The elasticity of the materials of bearing and journal;
5. Bearings not properly supported;
6. Insufficient supply of oil when starting;
7. Insufficient supply of oil after starting, which will depend upon (a) method of supply, and (b) where oil is introduced to the bearing.

It is thus evident that even with perfect lubrication, the fitting of the bearing must not be neglected, as an ill fit will produce wear even after the oil film is formed and shorten the life of the brasses. This wear will be further reduced by keeping the journal from touching the brass as little as possible when starting. This can be affected by having an abundant supply of oil at each bearing at the instant of starting.

III.—*Methods of Lubrication.*

As previously stated, the ideal to be attained in lubrication is that condition where the film of lubricant is maintained between the rubbing surfaces under any load that the machine may be forced to carry.

In practice this ideal is not very often thought of, too much attention being given to saving oil and to installing the lubrication system where it is most convenient to do so from the standpoint of accessibility. There are, however, two general

systems of lubrication in use: (1) Wick lubrication; (2) bath lubrication.

Wick lubrication is often nothing but lubrication with a limited supply of oil, the disadvantages of which have been previously pointed out. Under this head may be placed any forms of automatic lubrication which give the supply of oil to the bearing surfaces by drops or in a limited supply. This method should obviously not be used where it can be avoided, unless the supply is more than sufficient for the bearing surfaces, and the surplus oil can be collected and used over again.

Bath lubrication is primarily that system where the journal turns in a bath of oil and can at all times obtain all the oil it is able to mechanically carry between the journal and brass to form the oil film. Under this head may be placed efficient ring lubrication, forced lubrication, and gravity-feed lubrication. Forced lubrication is that system in which the oil is supplied by pump to the bearing under a low pressure. By this system the bearing surfaces are assured at all times of an abundant supply of oil; the surplus oil running from the bearing and collecting in a well under the machinery, is used over again. Gravity-feed lubrication is nothing more than forced lubrication, the pressure at which the oil is delivered at the bearing being due to the head. The advantage of forced lubrication is that it not only insures an abundant supply of oil, but in rapidly-moving reciprocating parts the oil is forced entirely in when the pressure is released on one-half of the bearing, and completely fills the space between brass and journal, whereas by wick lubrication this space would be only partially filled due to the rapidity of revolution and consequently the short time during which the pressure is relieved on half the bearing.

The advantages of using bath lubrication in any of its forms on any machinery may be summed up as follows:

1. Reduction of power lost;
2. Reduction of wear of bearing;
3. Reduction of time necessary for adjustment;

4. Reduction of time necessary for renewal of bearing;
5. Reduction of cost of maintaining machinery;
6. Reduction to minimum of chance of bearing heating and consequent stoppage of machinery at critical times;
7. Reduction in attendance on machinery.

Prof. John E. Sweet says: "Reduction of friction of necessity implies an additional saving in maintenance, attendance, repairs and, above all, *saving loss occasioned by delays.*"*

It is, of course, assumed that means are provided for collecting the surplus oil, filtering it, freeing it of water and using it over again. This feature can be used with any system of lubrication and will result in reduction of amount of oil used.

The point at which oil is introduced to a bearing should be arrived at by considering (first) in what direction the load acts, and (second) whether motion is reciprocating or not.

If the load is constant in direction, such as would be exhibited on a line shaft where the belting is pulling in one direction or nearly in one direction, or in a turbine where the weight of the turbine is acting on the bottom half of the bearing, then the oil should be introduced at the point of minimum pressure (see Fig. 12) or at a point almost at right angles to the line of load. In this way the full effect of the mechanical action of the revolving shaft is utilized to form and maintain the oil film if the supply of oil is abundant.

In the case of reciprocating parts, where the load comes alternately on one and then the other half of the bearing, the oil should be introduced to take advantage of (1) the mechanical action of the revolving journal to form and maintain the oil film; and of (2) the release of pressure alternately on one and then the other half of the bearing. For this reason the oil should be introduced into two small chambers cut in the bearing metal and extending nearly the whole length of the bearing, one each on the on-side of each half of the bearing. This chamber if kept full of oil will give the bearing

* Author's italics.

full advantage of its film-forming properties. A circumferential chamber should be cut in the bearing metal extending around the middle of the bearing, each half connected to one of above chambers. This chamber practically divides the bearing into two bearings, and, if kept full of oil, will allow

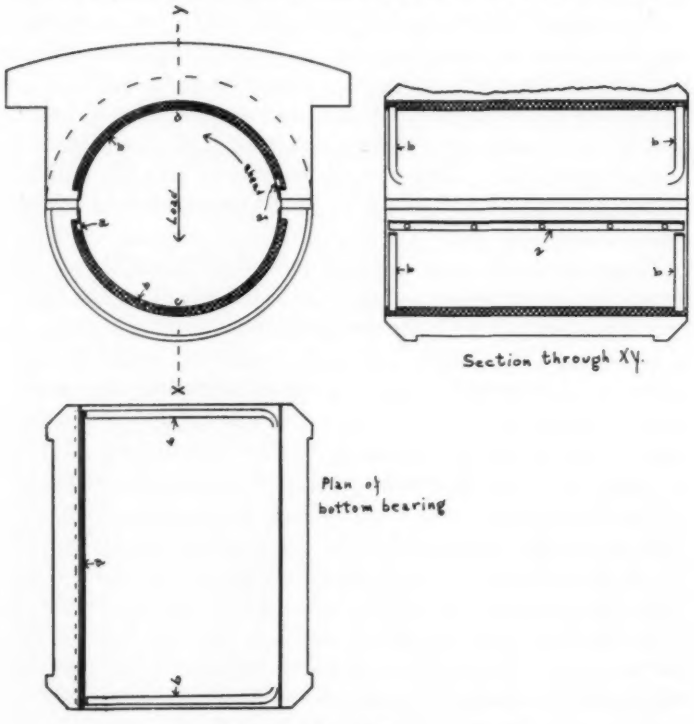


Fig. 17.

the bearing to take advantage of the alternate release of pressure, the oil running from the chamber into the space between the bearing and journal when the pressure is released. To facilitate this action in high-speed machinery, oil-ways may be cut longitudinally from this chamber almost to the ends of the bearing.

For either case where the load is constant in direction, or is alternating on one-half of the bearing and then on the other, either gravity or forced lubrication may be used; but with high speeds forced lubrication is preferable, as it is more positive in its action. In either case arrangements should be installed for collecting the surplus oil from each bearing.

In carrying out these ideas of oil supply, the bearings shown in Fig. 17 and Fig. 18 have been designed. Fig. 17 is a

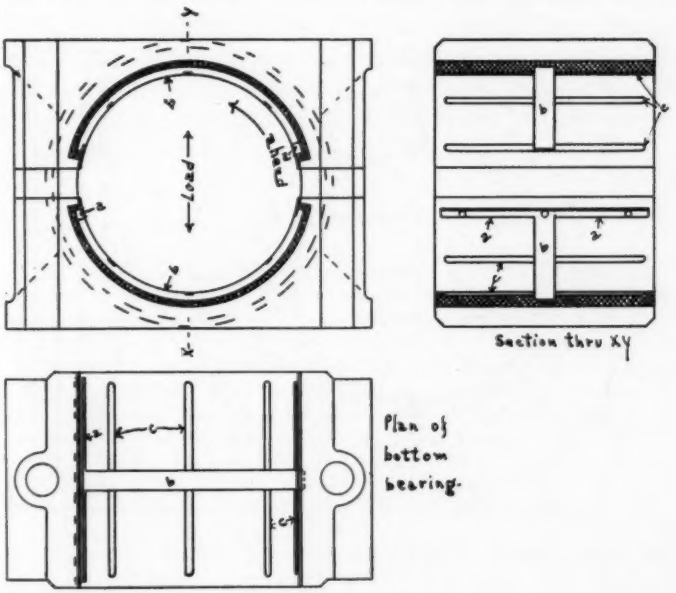


Fig. 18.

bearing where the load is constant in direction, such as a line-shaft bearing or a turbine bearing. The oil may be supplied to the spaces, *a a'*, by either forced or gravity feed direct from the oil tank or from a sight feed. The oil-ways, *b*, cut near the edges, are for the purpose of collecting part of the oil that would run out of the ends of the bearing, and of allowing it to be carried around to the opposite side of the bearing. The oil film in this case will be maintained under

both *c* and *d* by the mechanical action of the rotating journal; the journal constantly carrying oil from *a* to *c* and from a_1 to *d;* and when motion is reversed, from *a* to *d* and from a_1 to *c*. No other oil-ways should be cut.

Fig. 18 is a bearing where the load alternates on either half of the bearing. Oil is introduced at *a* either by forced lubrication or gravity; the spaces *a* and *b* should always be kept full of oil in either case. Where the oil is allowed to flow into

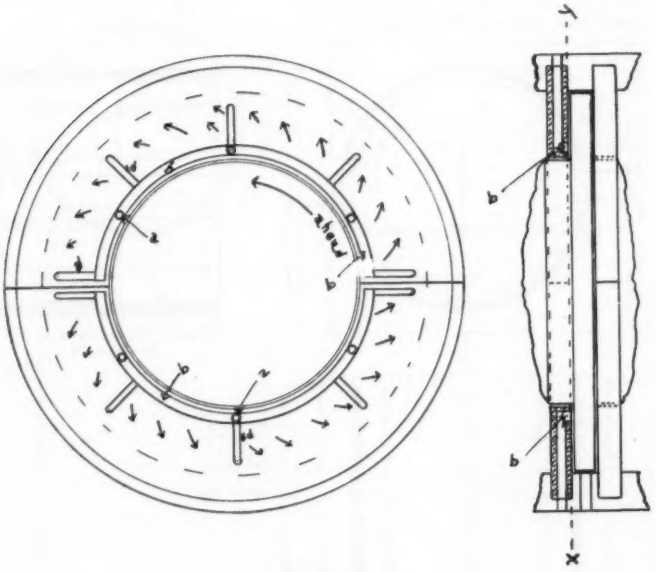


Fig. 19.

a and *b* in sufficient quantity to keep these spaces full but not under pressure, the effect would be similar to simple bath lubrication, and in order to get the oil quickly to the rubbing surfaces the oil-ways *c* might be cut. The space *a* being kept full of oil, the journal is allowed to carry the oil around with it and form the oil film as in Fig. 17, and the space *b* being kept full of oil, the oil runs in between the bearing and journal longitudinally when the pressure is relieved on either half of the bearing.

In the case of thrust bearings the oil will tend to flow radially, due to centrifugal force, and will also tend to flow circumferentially, due to the motion of the thrust collar. The oil should then be introduced as near the inner edge of the thrust bearing as is possible, at several points of the bearing, the number depending on the size of the shaft. Fig. 19 shows a thrust bearing designed to effect the result of keeping a film of oil constantly interposed between the thrust collar and bearing. Oil enters at the points *a* and fills the chamber *b*, and as the shaft revolves the oil works out toward the outside circumference of the bearing in the direction indicated by the arrows. In large thrust bearings, oil grooves, *d*, could be cut to facilitate the distribution of oil, but these should not be necessary in small thrust bearings.

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NOTES ON METALLURGY AND HEAT TREATMENT OF IRON AND STEEL.

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This paper is a compilation, in a condensed form, of some of the essential facts relating to metallurgy collected from various text-books and from observations by the writer during the summer of 1910.

Iron and steel enter into almost every branch of industry and are naturally most essential to us; they form the largest manufactured product in the world. Steel, wrought iron and cast iron form what is known as the "ferrous group" of metals, *i.e.*, what is classed as iron and steel. Cast iron and steel have two characteristics in common: first, they each contain at least 92 per cent. of iron; and, second, carbon is their next most important element. It is this element which controls and regulates their chief qualities. Cast iron is produced in final form in almost every city of any size in the United States and hardly less generally in other civilized nations. The manufacture of steel is more or less centralized for economical reasons, and is several times as great as cast iron in volume. Wrought iron is made less generally, and in less amounts than either cast iron or steel, but it has its uses and importance.

The raw material of all ferrous products is iron ore. The most important ores are the oxides, which are simply natural iron rust, with iron and oxygen in combination. If the oxygen can be gotten rid of the iron remains. If charcoal be burned and iron ore be added, the combustion being assisted by a hand bellows or other blast, the iron will separate into pasty globules and the oxygen of the ore will unite with the

carbon and pass off as a gas. The temperature reached by this process is not sufficient to melt the iron nor is it high enough to cause the iron to absorb any considerable carbon, thereby forming pig iron; but the temperature is high enough to make the pasty globules weld or stick together. In this way, for thousands of years, iron was made the world over. With development the height of the furnace and the force of the blast were increased and, with the resulting higher temperature attained, modern pig iron, from which the steels of to-day are made, was produced.

ORES OF IRON.

The most common ores of iron are: Magnetite, or magnetic oxide of iron, also called black oxide of iron (Fe_3O_4); haematite, red oxide of iron (Fe_2O_3); and siderite, or ferrous carbonate ($FeCO_3$).

Magnetite.—This is the richest of the ores of iron, and when pure contains 72.4 per cent. iron. The most characteristic form of this ore is known as "lodestone," from its magnetic properties. This occurs in great deposits in the United States, Norway, Sweden and Siberia. The magnetite ores of Sweden are the purest that are found in large quantities anywhere, and from these is made "Swedish Iron," famous throughout the world for its purity. The tool-steel and cutlery industries of Sheffield, England, are dependent on the Swedish product for their raw material. Most of the deposits of magnetite in the United States are either too poor in iron or contain too many impurities to warrant their being worked profitably.

Haematite.—These ores are found principally in the Lake Superior mining region, Iron Mountain in Missouri, in northern New York, Alabama, Germany, France and Belgium. The best haematite contains 70 per cent. iron; the Lake Superior deposits contain about 68 per cent. iron, while the hydrated haematites run lower, generally not yielding more than 50 per cent. iron. The haematite ores are the principal source of raw material for the iron industries of the United States.

Siderite.—This ore is very poor and is not mined to any considerable extent in the United States, but in Great Britain siderite furnishes about one half of all the iron made. This ore forms the famous "clay ironstone" of the Cleveland district in England. It runs about 48.2 per cent. iron when pure.

Iron Pyrites (FeS_2).—Iron pyrites is quite common, but it is so poor in iron and it is so difficult to remove the sulphur that it can not be profitably worked in competition with the richer ores.

About 40,000,000 tons of haematite-iron ore are mined yearly in the Lake Superior region; New York and Alabama each produce over one million tons. There are imported from Cuba and Spain a total of about 1,000,000 tons yearly of ore of various kinds, which is used generally in blast furnaces on the Atlantic seaboard and for mixing with other ores at various works inland.

CHARACTERISTICS OF FERROUS PRODUCTS.

Steel pipe, merchant bar, etc., made of low-carbon steel are very often sold under the name of wrought iron, and high-carbon steel masquerades as crucible steel; it is therefore considered advisable to cite the following definitions:

Cast Iron.—Iron containing so much carbon or its equivalent as not to be malleable at any temperature. The dividing line between cast iron and high-carbon steel is placed arbitrarily at 2.2 per cent. carbon. Any ferrous product having more than this percentage is classed as cast iron.

Pig Iron.—Cast iron which has been cast into pigs at a blast furnace and is then ready for use in cupolas or in furnaces for steel making. This term is also sometimes applied to the molten cast iron that is ready for pouring into pigs before it has been cast into any shape.

Spiegeleisen.—Pig that runs very high in manganese.

Malleable Cast Iron.—Cast iron which is first molded into the shape desired and is then made malleable by subsequent treatment without being fused.

Malleable Iron.—This is the English name for *wrought iron.* When used in the United States the term signifies *malleable cast iron.*

Wrought Iron.—An iron practically free from carbon, but bearing slag as an impurity. It does not harden materially when suddenly cooled, and cannot be tempered. It cannot be melted and cast in a mold, but is always forged to the desired shape and size. Wrought iron of to-day is generally made by the puddling process, and usually contains less than 0.12 per cent. carbon; when the amount of carbon is above 0.30 per cent. the metal is then called "Weld Steel," sometimes "Wrought Steel."

Steel.—Steel includes all impure irons which, in consequence of the presence of other elements, have the property of hardening when suddenly cooled from a high temperature, and of taking a definite "temper," or degree of hardness, by a definite modification of temperature, and which may also be forged.

The Boundary Between Wrought Iron and Steel is most difficult to define. Some authorities draw the line at 0.2 per cent. carbon, while others place it as high as 0.32 per cent. carbon.

Steel making may be divided into three classes:

1. By carburizing or carburetting malleable or wrought iron, as in the crucible process.

2. By partially decarbonizing metal rich in carbon, as in one form of the Bessemer process, and in the puddling process.

3. By completely decarbonizing high carburetted iron and then recarbonizing to the desired degree in a single process, as in the "Bessemer" and the "Siemens-Martin" processes.

The details of these processes are readily obtainable and will not be described here.

Chemically and absolutely pure iron is probably an unknown substance. Nearly pure iron is made by several methods, or it may be deposited by electrolysis. In its almost pure state it is a silvery white and very lustrous metal; it is soft, ductile and malleable; it is tough though not remarkably strong; it is heavy, very easily oxidized, and an excellent conductor of

heat and electricity. The purer iron is the more strongly it can be magnetized, but the magnetism is more permanent in proportion as the iron contains more carbon or other steel-making elements alloyed with it.

Crucible steel is the most costly of all steel, its price being about three times that of the next best, *i.e.*, acid open-hearth steel. The general reason for its excellence is believed to be that it is made in a vessel free from air and furnace gases and practically devoid, or nearly so, of oxygen, nitrogen and hydrogen. Perhaps the high quality is largely due to its being made in small quantities, where it is under better control and gets better attention.

Since wrought iron usually costs about 20 per cent. more than ordinary low-carbon steel, there is a great tendency for the dealer to substitute low-carbon steel for it. The superiority claimed for wrought iron over soft steel is that it is purer and contains slag; the presence of slag tends to increase the metal's toughness and resistance to breaking when subjected to bending or to a sudden blow or shock. It is a mistaken idea that the slag improves weldability, as the low carbon content seems responsible for this quality. Since wrought iron contains slag, it may thus be distinguished from low-carbon steel. Generally the percentage of slag in steel runs about 0.2 and less, while in wrought iron there is 1 per cent. or more. Normal wrought iron is free from manganese, while Bessemer and open-hearth steel contain about one-half of one per cent., or more. On the other hand, phosphorus is generally present in *wrought iron* to the extent of more than 0.1 per cent., while *good steel* should never contain that much. Crucible steel usually contains more carbon than Bessemer or open-hearth, it ordinarily has less than 0.4 per cent. manganese and 0.2 per cent. silicon; while open-hearth and Bessemer normally have more than 0.4 per cent. manganese and less than 0.2 per cent. silicon. This comparison does not hold, however, with castings, as the silicon content there sometimes runs up as high as 0.6 per cent. with both open-hearth and Bessemer steels. Both Bessemer and open-hearth steels can be made low in man-

ganese, but the manganese and silicon cannot both be low without producing blowholes.

Acid and basic open-hearth steels may be distinguished from each other by there being usually more silicon and phosphorus and less manganese in the acid product. This same distinction holds with acid and basic Bessemer.

Basic open-hearth and basic Bessemer differ in the fact that the former is lower in manganese, silicon and phosphorus, and also that it dissolves more slowly in dilute hydrochloric acid than basic Bessemer does.

INFLUENCE OF THE ELEMENTS IN CAST IRON.

The effect produced in iron by the various elements generally found therein is determined not only by the character of the elements themselves but by their mutual interactions. This fact of the mutual interactions must not be lost sight of as it is most important.

Carbon.—In cast iron this is the most important element, and it is due to the presence of carbon that the metal can be melted easily and made into castings. The melting point of cast iron is determined by the amount of total carbon, and the degree of hardness and the strength imparted to iron castings are also dependent on it. Carbon appears in iron in either the "combined" or the "graphitic" state. About 3.8 per cent. carbon is as much as cast iron can hold in the combined state; above that the excess is graphitic carbon. All the carbon content is supposed to be combined with or dissolved with the iron while the iron is molten, and when the iron becomes solid the carbon will remain in the combined state unless there be some influence to prevent it. Iron containing only combined carbon is white and very hard. Slow cooling of a casting will cause some of the carbon to be of the graphitic form. Should the metal be cooled very suddenly, so that the carbon cannot assume the graphitic form, the casting will be white, but should the iron contain more carbon than it can hold in the combined

state the darkest casting will result from that which contained the most carbon to begin with.

As a summary of the effect of carbon in cast iron it may be stated that cast irons become harder as the proportion of combined carbon increases, but as soon as a certain proportion is passed the influence of graphitic carbon is felt and the iron becomes softer and weaker until, when containing 5 per cent. or 6 per cent. carbon, one-half of which is often in the graphitic state, the iron becomes very soft and easily cut, of low density and of little strength.

Silicon.—Silicon is a softener when added to hard cast iron. With from 2 to 3.5 per cent. silicon the carbon seems to be driven out of its combined state and is changed into graphite. The iron remaining is softer, because iron itself is soft; the original iron was hardened by the carbon that was combined with it. Probably silicon, no matter how small the quantity, hardens the iron, but at the same time its effect in producing graphitic carbon is so great that the sum total of its effects is to soften the metal. The chilling tendency of iron is lessened by silicon.

Phosphorus.—Phosphorus produces fluidity, and when the content is more than 1.5 per cent. a brittle iron is the result. Cast iron containing phosphorus is liable to break under shock. Probably no other element of itself so weakens cast iron as does phosphorus, especially when it is present in large quantities. However, with a large proportion present, say from 2 to 5 per cent., the molten metal is very fluid, and thin, intricate castings are easily made; but it must be remembered that castings of these compositions cannot be used where great strength is desired. If a grey cast iron containing phosphorus be heated to a temperature of 900 degrees C., globules of iron rich in phosphorus ooze out from the mass, thus making the original mass honeycombed. Hence phosphoric irons are not suitable for articles which are subjected to high temperature, for example: ingot molds or grate bars.

Sulphur.—This element's influence upon the formation of graphite is almost the direct opposite of that of silicon, *i.e.*, each increase of sulphur increases the amount of combined

carbon. It neutralizes the effect of silicon; it is considered that 0.01 per cent. S. neutralizes 0.15 per cent. Si. in its effect upon the condition of the carbon in the iron.

Manganese.—Cast iron *when free from manganese* cannot hold more than 3.8 per cent. carbon in the "combined state," but with an increase in the manganese there is an increase in the combined carbon over this percentage; this is shown in spiegel iron (an iron rich in Mn.), where the combined carbon is 5 per cent., and with ferro-manganese, where the combined carbon runs as high as 7 per cent. When, however, manganese and sulphur are both present, the Mn. and the S. form MnS., which does not increase the proportion of carbon in the combined state. There is rather the reverse effect, because the FeS., in which form the sulphur is most powerful in producing combined carbon, is broken up. The excess Mn. over that used to form MnS. is useful in increasing the combined carbon. So we have the peculiar situation that if the manganese be high any increase in sulphur will decrease the tendency of the manganese to form combined carbon. We may make the paradoxical statement that manganese and sulphur both tend to increase the total carbon and the combined carbon, and yet they neutralize each other.

The following useful points about iron castings may be observed. Strength increases with increase of silicon up to 3 per cent. Silicon decreases the combined carbon and thus prevents brittleness and increases strength. Foundry experience seems to show that the lower the silicon the weaker the small castings and the stronger the large castings; and the higher the silicon, up to 3 per cent., the stronger the small castings and the weaker the large ones. Consequently, with large castings it is usual to use the least silicon that will give the requisite degree of softness.

The center of a casting is always coarser grained and weaker than the surface. Removing the skin of a casting reduces its strength per unit of cross section. Bars cast horizontally are more uniform in strength and size than those cast vertically. Stronger castings are made in green-sand than in dry-sand

molds, because castings cool slower in dry sand. Square test bars are stronger than round test bars of the same area of cross section. With a square test bar the relative tensile strengths of the corner, the side and the middle of the bar are respectively, 16.4, 15.4 and 13.7 tons. The corner is thus seen to be the strongest, the side the next, and the middle the weakest. Rumbling castings increases their strength.

As a resumé of the effects of the elements herein stated, it is seen that it is the combined carbon which is the great hardener of cast iron. The other elements produce hardness to a greater or less degree, accordingly as they produce combined carbon, with the exception that manganese not only produces combined carbon but also produces $(FeMn)_3C$., a compound which is very hard and difficult to machine. Silicon, by decreasing combined carbon, diminishes hardness, but when there is a silicon content of more than 3 per cent. there are new compounds formed with the silicon which make the iron very hard. More than 3 per cent. silicon also increases combined carbon. The greatest softness in cast iron is obtained with from about 2.5 to 3 per cent. silicon, with not more than 0.1 per cent. sulphur and with manganese not over 0.4 per cent. With large slowly-cooled castings the silicon should be nearer 2.5 per cent., while with small quickly-cooled ones it should be nearer 3 per cent., so that the combined carbon shall not be over 0.15 per cent. and the graphite not over 3 per cent. This mixture would be very soft and would machine easily, its transverse strength being about 2,000 to 2,200 pounds, and its tensile strength being about 22,000 pounds per square inch. There are two ways to increase the strength of this mixture without increasing the hardness; the better way is to cut the sulphur and phosphorus down as low as possible, because these two elements are the ones which weaken cast iron the most; the other way is to decrease the total carbon, and hence the graphite, for graphite crystals, especially when large, are great weakeners of iron. The smaller the crystals of graphite in iron, the stronger the iron. The total carbon may be decreased by melting steel scrap with the iron or by decreasing

the manganese, but it must be observed that the manganese content must always be twice that of the sulphur or else the iron will be brittle and weak. The strength of cast iron may be increased by increasing the combined carbon, which is done at the expense of workability and softness. Cast iron with from 1.5 to 2 per cent. silicon, 0.9 per cent. combined carbon, 0.5 per cent. Mn., not more than 0.08 per cent. S. and 0.3 per cent. P., will be workable without difficulty and at the same time have a tensile strength of more than 28,000 pounds per square inch. Should the obtaining of low P. and S. entail too great expense the necessary strength may be obtained by increasing the Mn., but here trouble will be met with again as the increase of Mn. decreases the softness and causes high total carbon.

WROUGHT IRON.

In the production of wrought iron from pig, the pig being melted on a hearth of iron ore, the silicon, manganese, sulphur and phosphorus unite with the oxygen to form slag, while the carbon passes off with the oxygen as CO_2 and CO. The heat of an ordinary reverberatory furnace is not sufficient to maintain the mass in a molten state. It becomes viscous, then pasty, and is worked into balls, removed from the furnace, and either put through the squeezer or hammered into a bloom. The puddle ball consists of numerous globules of almost pure iron with slag in the interstices between them. Much of the slag is expelled by repeated rolling through the squeezer, but it is practically impossible to get rid of the slag or cinder entirely, and the finished bar is permeated by it, thus forming planes of separation between the metallic iron. These little films of cinder weaken the structure by destroying the cohesion of its particles, but in another way they materially aid the strength of the product, for the sulphur and phosphorus are never entirely removed during puddling, and there is always sufficient quantity of these elements left to produce bad results if they are so combined as to exert the full effect caused by their crystallization—the network of slag prevents their ten-

dency to crystallize. There are five elements found in pig iron which have a material effect upon the finished product—*wrought iron*. These are: silicon, manganese, carbon, sulphur and phosphorus. The influence of these elements upon wrought iron has never been fully determined, owing to the many disturbing conditions, foremost among which is the effect of varying amounts of work upon the finished material. The same variation also arises with steel, but it is more important in the case of wrought iron, since the strength of wrought iron is materially influenced by the thoroughness with which the globules have been welded together. Phosphorus and silicon harden wrought iron, as does carbon. Phosphorus up to 0.2 per cent. does not harm, but probably improves, iron containing silicon not above 0.15 per cent. and carbon not more than 0.3 per cent. None of these elements except carbon seems to aid the welding property. Sulphur produces brittleness at high temperatures, or, as the blacksmith expresses it, makes iron "hot-short."

STEEL, INFLUENCE OF THE ELEMENTS.

Carbon.—This element exercises the most important effect in steel. It increases strength and hardness, and reduces ductility. Starting with a tensile strength of 40,000 pounds per square inch for low-carbon steel, each increase of .01 per cent. in carbon gives an increase in tensile strength per square inch of about 1,000 pounds for acid, and 770 pounds with basic open-hearth steel. If a piece of steel be heated to redness and plunged into water it becomes very hard; this is due to a peculiar union between the carbon and the iron. Should the carbon content be 0.3 per cent. the steel so quenched will be hard enough to scratch glass, should the carbon be as high as 0.6 per cent. the steel is suitable for ordinary cutting tools. With 1 per cent. carbon the steel about reaches its limit of hardness.

Silicon.—Silicon is of use in the manufacture of steel by the Bessemer and other pneumatic processes, it being the fuel that

is used to raise the temperature of the molten metal in the converter to a higher point than could have been obtained by the combustion of the carbon alone. Its greatest value is that its presence prevents blow holes. It is also a hardener of steel, its hardening power being similar to that of carbon, 0.5 per cent. is about the maximum of silicon that occurs in steel; this percentage is used in steel castings. In structural steel there is seldom more than a trace of this element.

Manganese.—Manganese increases tensile strength only when the percentage of Mn. is over 0.3 or 0.4 per cent. With less percentage the effect of manganese is probably masked by the presence of some other element. It gives ductility to the steel, especially while hot, and gives ease of working. It increases hardness, and especially that kind of hardness which makes it extremely difficult to machine the metal. With more than 1 per cent. of manganese the metal is very brittle, and when the content is above 2 per cent. the steel becomes so brittle as to be practically worthless; but should the percentage of Mn. be as great as 7 per cent. there is a reversal of its influence, and the metal is then not only extremely hard and practically impossible to machine, but upon being subjected to certain heat treatments becomes ductile and tough. Manganese is used as an antidote to sulphur.

Sulphur.—This element causes brittleness at high temperatures; its effect is mostly felt in the absence of other impurities. With cold metal its effect seems to be unimportant. It also seems to prevent welding; steel or iron containing above 0.2 per cent. sulphur being welded with difficulty.

Phosphorus.—This is the most undesirable of all the elements found in steel. It has very little effect upon the hot properties of steel, but in the cold state it makes the metal brittle and adds to the tensile strength in about the same degree as carbon does. In ordinary steels the phosphorus is always limited to 0.1 per cent. and with special steels it is even less. For a metal that is to stand vibratory stresses and sudden shock phosphorus is probably the most dangerous of the elements which can be used to increase strength; for an

increase of strength secured by this means is accompanied by a much greater decrease in ductility than if the same increase in strength were obtained by using carbon.

As a summation, the following conclusions may be drawn concerning the strength-producing qualities of the elements in steel:

Carbon.—Each 0.01 per cent. of carbon in acid steel strengthens the metal by 1,000 pounds per square inch, when the carbon content is determined by combustion analysis. The effect in increasing strength is 1,140 pounds for every 0.01 per cent. carbon, as determined by color test. This discrepancy is due to the fact that the color test does not determine all the carbon present. With basic steel, every 0.01 per cent. carbon strengthens steel 770 pounds per square inch by combustion test, and 820 pounds per square inch by color test.

Phosphorus.—Each 0.01 per cent. strengthens steel 1,000 pounds per square inch.

Manganese.—Each 0.01 per cent. of Mn. gives a strengthening effect which increases as the carbon content is greater. In steels with less than a certain content of manganese the effect of the Mn. is complicated, due to some disturbing condition, probably iron oxide, so that a decrease in Mn. in very low-carbon steels gives increased strength. With acid steel each increase of 0.01 per cent. above 0.4 per cent. Mn. raises the strength an amount varying from 80 pounds with 0.1 per cent carbon to 400 pounds per square inch with 0.4 per cent. carbon. With basic steel each increase above 0.3 per cent. Mn. increases the strength an amount varying from 130 pounds per square inch in a metal having 0.1 per cent. C. to 250 pounds in a metal with 0.4 per cent. carbon.

Sulphur.—The effect of this element upon the strength of acid or basic steel is practically nil.

ALLOY STEELS.

Ordinary steel, in order to distinguish it from Alloy Steel, is known as *Carbon Steel.* It is considered as an alloy of carbon and iron. Alloy steels comprise those steels to which a

controlling amount of some alloying element in addition to the carbon is added. The international committee on the nomenclature of iron and steel defines alloy steels as follows: "Those steels which owe their properties chiefly to the presence of an element (or elements) other than carbon." The distinction between the addition of an element to produce a slight benefit to carbon steel and one added to produce an alloy steel is sometimes a very delicate one. For example, manganese is commonly added in amounts less than 1.5 per cent. to most open-hearth and Bessemer steels to get rid of oxygen and to counterbalance the effect of sulphur. Similarly, silicon is added in small quantities, 0.1 to 0.2 per cent., to give sound ingots; but in neither of these cases is the product called alloy steel.

But when a steel containing 10 to 20 per cent. manganese or 2 to 3 per cent. silicon is made, the resultant is a steel with entirely different qualities from that made without the addition of either the Mn. or Si., and an alloy steel is obtained. Before proceeding, it may be well to define what is meant by "Ternary" and "Quaternary" alloys. An alloy containing iron, carbon and one other influential element is called a "ternary alloy" or "three-part" alloy. The most common alloy steels under this classification are: Chrome steel, nickel steel, manganese steel, tungsten steel, silicon steel, vanadium steel and titanium steel. Should the steel contain iron, carbon and two other influential or alloying elements the alloy is called "quaternary." The commonest examples under this head are: nickel-chrome-steel, tungsten-manganese, tungsten-chromium, nickel-manganese, manganese-silicon, tungsten-molybdenum, tungsten-nickel and nickel-vanadium steels. The effect produced by adding an alloying substance to carbon steel is most surprising and often incapable of being foretold, and that resulting from a combination of two alloying substances is even more so. New products result, with properties entirely different and oftentimes almost the opposite of those produced by its individual constituents.

The manufacture of alloy steels is a very simple operation.

Generally the alloying substance is added like the recarburizer, *i.e.*, in the ladle, as is the case with manganese steel, where the necessary amount of ferro-manganese is added at the end of the operation. With nickel steel the ferro-nickel may be added in the same way, but most commonly shotted nickel is added in the furnace and allowed to dissolve in the bath of molten metal before tapping. Tungsten and tungsten-chrome steels are generally made by the crucible process; the requisite amount of ferro-tungsten, or metallic chromium and metallic tungsten, being placed on top of the charge when filling the crucible. While the manufacture of alloy steels is very simple, their treatment is more complicated. Nickel steel can be heated, rolled or forged with little difficulty, but with manganese, tungsten and some of the other steels this treatment is very much more difficult.

Nickel Steel.—This is the most widely used of all the alloy steels. The nickel varies from 1.5 to 4.5 per cent., but most usually it is from 3 to 3.75 per cent., and the carbon about 0.2 to 0.5 per cent. (most generally about .25 per cent.). This does not include armor plate, since that is a quaternary alloy, containing nickel and chromium. Nickel steel is mostly used for locomotive axles, engine forgings, shafting, frame and engine parts for automobiles, as well as automobile axles, etc.

Nickel steel has a higher elastic limit than carbon steel, and the higher elastic limit is obtained at very little reduction in ductility; for example, 3 per cent. nickel added to 0.25 per cent. carbon steel gives a metal as strong as simple carbon steel of 0.45 per cent. carbon, but with the ductility of carbon steel of 0.25 per cent. carbon. It is this increase in elastic limit that gives to nickel steel the property of standing fatigue (*i.e.*, the power of resistance to repeated stresses and to alternating stresses), under which all steel will ultimately break down, even though the load is far less than that which it can bear indefinitely if the load be constantly applied. By repeated stresses is meant stresses put upon a body at intervals and relieved meanwhile; while alternate stresses are stresses first of compression and then of tension, as the stresses in a wire

that is bent back and forth, or in a rotating shaft that is out of alignment. With low-carbon steels, not annealed, each 1 per cent. of nickel up to 5 per cent. gives an increase of 5,000 pounds per square inch in elastic limit and about 4,000 pounds per square inch in ultimate tensile strength. With nickel steel containing 3.5 per cent. nickel the elastic limit should be about 60 per cent. the ultimate strength; while with ordinary carbon steel the elastic limit is only about one-half the ultimate strength. The modulus of elasticity of nickel steel containing about 4 or 5 per cent. nickel is about the same as that of carbon steel, it being about 29,500,000 pounds; however, with a higher per cent. of nickel, especially with more than 20 per cent., the modulus of elasticity is lower. This results in the steel being more resilient or springy, hence the reason for not more than 4 per cent. nickel being put in steels for structural use, because a bridge built of steel that is resilient, even though the steel be strong enough, would vibrate excessively with a passing load and might, perhaps, even be unsafe on account of the repeated stresses set up. Nickel steel is harder than carbon steel, but not so much so that it cannot be machined without difficulty; this advantage is made use of in employing nickel-steel rails for curves and other places where ordinary steels soon wear our. Nickel steel has superior stiffness, and yet bends to a greater angle before rupture than does carbon steel; and plates of nickel steel are not weakened as much by punching as are those of carbon steel. This alloy steel possesses greater compressive strength and greater toughness under impact as well as greater shearing strength than carbon steel; the latter property is the reason why nickel-steel rivets are now specified for structural uses, because, on account of the greater strength of nickel steel, smaller rivets may be used, and this means smaller rivet holes in the structural members, thus leaving more of the structural steel to support the strains. However, since nickel steel does not weld as easily as carbon steel, greater care must be exercised in upsetting the rivets. Another great advantage in using nickel steel is that castings made of this material are relatively free from blowholes; the

melting point of nickel steel is also lower than that of carbon steel.

One peculiarity of nickel-steel alloys with less than 25 per cent. nickel is that their critical changes are irreversible. By this is meant that a change which takes place at a given temperature on cooling is not reversed on heating to the same or anywhere near the same temperature. That is to say, should a 20 per cent. alloy of nickel steel be cooled, the solid solution is still intact, and the "alpha" allotropic modification is not assumed until a temperature of below 100 degrees C. is reached. This steel having thus been cooled to 100 degrees C. and the solid solution having been decomposed into the alpha modification of the alloy, it is necessary to heat the metal to 600 degrees C. before the reverse condition takes place and the solid solution is again formed. When the percentage of nickel is 40 per cent. or above the critical transformations are reversible like ordinary steels, *i.e.*, they occur at very nearly the same temperature on heating as the reverse change does on cooling. It is interesting to note that the greatest tensile strength and elastic limit of nickel steels occur with those steels where the characteristic of irreversibility is most pronounced, *i.e.*, in steels containing between 12 and 25 per cent. of nickel.

Nickel steel has a very small coefficient of contraction and expansion, and when the amount of nickel is 36 per cent. the coefficient is less than that of any other alloy or metal known, it being practically zero. This alloy is patented and sold under the name of "*Invar.*" It is used in the manufacture of scientific instruments, pendulums of clocks, steel tape measures for accurate work, etc. When the proportion of nickel reaches 42 per cent. the steel has the same coefficient of expansion as glass at the atmospheric temperature. Since platinum was heretofore the only known substance possessing this quality, the alloy has very aptly been named Platinite. It is used in the manufacture of armored glass, and for the electric connections passing through the glass plugs in the base of incandescent electric lights.

MANGANESE STEEL.

The discovery of this steel was due to the untiring efforts of Mr. Hadfield, of Sheffield, England, and it is most singular to note that the properties of this alloy are quite different from, in fact they are almost the opposite of, what might presumably be expected. When the manganese proportion is over 1 per cent. the metal is hard and a little brittle and practically worthless; with increase of manganese these qualities increase until, with a Mn. content of between 4 and 5 per cent., the steel is easily pulverized under a hammer. But on increasing the manganese above this point hardness and brittleness do not increase, and when a content of 7 per cent. manganese is secured an entirely new set of properties is developed. Manganese steel contains about 12 to 15 per cent. manganese and 1.25 to 2 per cent. carbon. This alloy has the very peculiar property of becoming very much tougher after quenching in water, without any great change in hardness. It possesses great strength and hardness in conjunction with ductility; its elastic limit is low, being about 35 per cent. of its ultimate strength. After manganese steel has been cast into an ingot or casting and is allowed to cool slowly it is almost as brittle as glass, but if it is then reheated to more than 1,000 degrees C. and rapidly cooled by being plunged into water, the formerly very brittle steel becomes as ductile as low-carbon steel, with a tensile strength about three times as great. The hardness of the steel remains about the same after quenching as it was when cast.

Manganese steel must be heated very slowly and uniformly, lest it crack. It is also difficult to forge, as forging must be done within a very narrow range of temperature above a red heat, and must begin with very light taps of the hammer. Manganese steel is used chiefly for rock-crushing machinery, railway crossings, railway frogs, mine-car wheels, burglar-proof safes and mining dredges; its use is restricted by the fact that it must be cast in the shape in which it is to be used, since machining it or cutting it to form is out of the question;

small changes in shape (as, for example, in making a fit) are accomplished by grinding.

Chrome Steel.—The characteristic of this alloy when in the hardened state is the combination of hardness with high elastic limit. Generally the carbon content is from 0.8 to 2 per cent. and the chromium runs from 1 to 2 per cent. Chrome steel is used for making armor-piercing projectiles and for very hard steel plate; this latter is not ordinarily used alone but is made into 3 and 5-ply plate, where a plate of wrought iron is sandwiched between two sheets of chrome steel. Should the hardened chrome plate be used alone the result would be a very brittle metal. The 3-ply plate is used for plows and also for safes. The 5-ply is employed for safes and jail bars; this consists of a central layer of wrought iron between two sheets of chrome steel, then two layers of wrought iron; the whole mass is welded together and hardened. The wrought iron gives ductility, and the chrome steel furnishes the hardness, so that the material cannot be drilled into or easily broken. "Krupp Armor" contains about 3.25 per cent. nickel, 1.5 per cent. chromium and .25 per cent. carbon.

Chromium steel containing about 2 per cent. chromium is used very widely for automobile purposes where great hardness is required, as gears, etc. More recently, however, a nickel-chrome steel is used; the nickel seems to give both an increase in the elastic limit and tensile strength.

Silicon Steel.—This steel, like manganese steel, was discovered by Hadfield. Before its discovery it was the general impression that pure iron possessed the highest magnetic force and permeability of any known substance or of any combination of substances that could be produced. However, silicon steel of the Hadfield composition and treatment has not only a greater magnetic permeability than pure iron but has a very high electrical resistance. Silicon steel is, therefore, very valuable in the manufacture of electro-magnets, and in electrical machinery is about the most efficient material known. Silicon steel made according to the Hadfield patent

covers a silicon content of from 1 to 5 per cent., but the alloy that he recommends contains 2.75 per cent. Si. and the smallest possible contents of carbon, manganese and other impurities. Before the steel is ready for use it is subjected to a double-heat treatment; first it is heated to between 900 and 1000 degrees C. and is cooled quickly, then it is reheated to between 700 and 850 degrees C. and allowed to cool very slowly, this second cooling process sometimes taking several days.

Vanadium Steel.—The manufacture of this steel is still in its infancy. Vanadium is, first, a great strengthener of steel; second, it is a great scavenger, in getting rid of oxygen, and nitrogen as well; it is claimed that it decreases segregation. In strength-producing qualities vanadium ranks next to carbon, and very wonderful results are secured by the addition of from 0.1 per cent. to 0.15 per cent., and more than 0.30 per cent. addition is not necessary. The great characteristic property of this alloy steel is its power of imparting great resistance to torsional stress and increasing the "life" of the metal. This steel will stand great vibratory stresses. In the manufacture of vanadium steel, the vanadium is not added to the bath until a couple of minutes after the manganese is put in for recarburizing, for if the vanadium is added first it is wasted greatly by oxidization. As is the case with all steels, it must not be forged at too high a temperature, and should be heated gradually; then, too, the forging at first should be a bit tender. This alloy steel is used for every purpose that will warrant the extra cost, which is about the same as that of 3.5 per cent. nickel steel. It is especially useful where lightness with strength is desired, as springs and axles of automobiles. It would seem that it possesses great advantages for shafts and other rotating parts.

Titanium Steel.—Titanium has not been used very much for making alloy steels, but its great affinity for nitrogen causes it to be highly recommended.

Self-Hardening Steels.—By this term is meant steels which will harden without being subjected to any heat treatment or

process to make them harden. It may also be defined as steel that cannot be softened or annealed by any known process; sometimes it is called "air-hardening steel." The critical temperature of such steels is below atmospheric temperature. All self-hardening steels are non-magnetic. The most extensive experiments with self-hardening steels were conducted by Messrs. F. W. Taylor and Maunsel White, during the time that they were associated with the Bethlehem Steel Works. The result of these experiments was the production of an entirely new kind of steel which has fairly revolutionized machine-shop practice. They discovered that by applying a new method of heat treatment to tools for machining metal very great toughness was imparted to the metal at a red heat and thus the possible cutting speed of the tools was surprisingly increased. So great were the cuts made with the tools that they became red hot due to the friction, and chips were removed that had a temperature of 300 degrees C. The steel did not lose its temper, its cutting edge nor its toughness at a red heat. The heat treatment that was employed consisted in heating the steel almost to the melting point and then plunging it into a bath of molten lead at a temperature of between 700 and 800 degrees C., where it was kept until it was the same temperature as the bath; the metal was then removed and plunged into oil and there cooled. This cooling was usually followed up by reheating the steel to a temperature between 370 degrees and 670 degrees C. It has been found that the heat treatment as applied by Messrs. Taylor and White is not necessary, the only requisite being to heat the steel to almost melting temperature and then cool it with moderate rapidity (in a blast of air, for example) until it is below a red heat.

The name of "high-speed steel" is a term very commonly applied to all steels having rapid cutting qualities. Soon it was found that it was the *tungsten* in the steel that gave the metal the properties of hardness and toughness at a red heat, and it was also learned that the presence of manganese and chromium in addition to the tungsten was unnecessary.

Larger percentages of tungsten were found to be desirable in order to make the steel more durable, and there are some steels with as high a proportion of tungsten as 24 per cent. The carbon at the same time is greatly reduced, it being from 0.4 to 0.8 per cent. with the best product. It is a mistaken idea that tungsten will take the place of carbon in producing hardness, as such is not the case. No amount of tungsten or any other element will make steel hard without the presence of carbon, or even when the carbon is low. The tungsten produces hardness by its effect upon the condition of the carbon, *i.e.*, by helping to retain the carbon in its solid solution, and not by any effect of its own. Ordinarily *"tungsten steel"* contains from 5 to 10 per cent. (and sometimes 24 per cent.) of tungsten and from 0.4 per cent. to 2 per cent. carbon. The normal composition of *Taylor-White* steel is:

	For cutting soft steel and cast iron. *per cent.*	For cutting hard steel. *per cent.*
Carbon	0.75 to 1.0	1.25
Chromium	3.	4.
Tungsten	8.5	8.5

The more recent "high-speed steels" are of the following composition:

Carbon	.4 to 2.19
Chromium	0. to 6.
Tungsten	3.44 to 24.
Silicon	.21 to 3.

At present the tendency seems to be to replace the chromium with manganese of from 1.25 to 4 per cent.

Molybdenum Steel.—Molybdenum is very often used in the manufacture of high-speed tool steel, it being used to replace the tungsten. It takes only about one-half as much molybdenum as it does tungsten to give the same results. In America the most successful percentages of molybdenum seem

to be from 6.0 to 15.0 per cent. High-speed steels can only be forged at temperatures above a bright-red heat, from 1,050 degrees to 1,150 degrees C. and higher.

Magnetic Steel.—Steel used for permanent magnets has about the same composition as high-speed tool steel. It contains generally about 4 to 5 per cent. of tungsten, and 5 to 7 per cent. of carbon. It is usually heated to a red heat (800 degrees C.) and quenched in water, and thus retains its magnetism better than ordinary carbon steel. Sometimes about 0.5 per cent. chromium is added to this alloy.

HEAT TREATMENT OF IRON AND STEEL.

In dealing with the changes which take place when iron and steel are subjected to the action of heat the concensus of opinion seems to be to follow the generalization adopted by Dr. Howe. By him all varieties of iron or steel are considered alloys of iron and carbon. The normal constituents of annealed steel are ferrite, cementite and pearlite.

Ferrite is theoretically pure iron, and especially iron free from carbon. It is magnetic, very soft and ductile, but is relatively very weak, having a tensile strength of about 45,000 pounds per square inch. It is an important constituent of all steels, and is the most important one in low-carbon steels. Wrought iron very nearly approaches ferrite. Under the microscope it is distinguished from cementite by its softness. Should a piece of steel containing these two be polished on a chamois skin stretched over a soft background, as a piece of wood, the ferrite will wear away and show intaglio. It may also be distinguished from cementite by subjecting a piece of steel to a 2 per cent. solution of HNO_3, or tincture of iodine; the ferrite will appear in darker grains and the cementite will be seen in bright thin flakes unattacked by the reagents.

Cementite is a definite carbide of iron (Fe_3C), it contains 6.67 per cent. carbon, is very brittle, harder than hardened steel, will scratch glass with ease, and is magnetic. In slowly-cooled steel, almost all the carbon present is in the form of

cementite. It is difficult for the unskilled eye to distinguish between a steel containing pearlite with a slight excess of cementite alone and a steel containing pearlite with a slight excess of ferrite. However, a chemical analysis settles the point, since with a carbon content of 0.9 per cent. or less there will be an excess of ferrite over pearlite, and with steels containing more than 0.9 per cent. carbon there will be excess cementite.

Pearlite.—In its most easily recognizable form pearlite consists of alternate layers of cementite and ferrite, the ferrite being the white streaks, and the dark streaks are cementite. Sometimes it separates into granules, and is then known as granular pearlite.

The softest and purest wrought iron consists almost entirely of ferrite with a little pearlite. As the percentage of carbon increases the ferrite is more and more replaced by pearlite, until with 0.9 per cent. carbon the ferrite is entirely replaced by pearlite. With a further increase of carbon content the pearlite is partially replaced by cementite, until with about 2 per cent. carbon the steel becomes *white cast iron.* Mottled cast iron may be considered as being made up of a bed of steel with graphite scattered through it; grey cast iron, as mild steel with graphite scattered through it, with certain irregularities in structure due to the presence of manganese, silicon, etc.

Should a piece of 0.50 per cent. carbon steel be allowed to cool slowly from a high temperature, when a temperature of about 700 degrees C. (1,392 degrees F.) is reached there is an interruption of the regularity in cooling. In some cases the cooling may be very slow, in some the bar may not decrease in temperature at all, while with some others there may be a decided increase in temperature for a moment, in spite of the fact that the steel is free to radiate its heat and that it has been cooling regularly down to that temperature. Moreover, it is found that when this "critical point" is passed the bar cools at a "uniformly retarded" rate until its temperature is that of the atmosphere. It must, however,

not be lost sight of that a bar cools more rapidly from 1,000 degrees C. to 900 degrees C. than it does from 200 degrees C. to 100 degrees C., and the term "uniformly retarded" is intended to cover that point. It is evident that there must be some change occurring in the metal itself that gives rise to the heat, and any point at which such an action takes place in steel is known as the "critical point." This point is called "A." The one taking place while retarding the temperature is known by "A" with the subscript "r"; thus: Ar. Now upon heating the same piece of steel the same changes take place at a temperature about 30 degrees C. higher than they do on cooling, the heat here being absorbed instead of given out. This point is also known as A, but has the subscript "c"; thus: Ac. Professor Howe has shown that in order to produce the change Ar the metal must be heated beyond Ac, and, likewise, in order for the change to take place at Ac it is necessary that the steel be first cooled below Ar.

In the case of a *pure* molten metal, say copper, the curve representing its fall in temperature is regular until the metal begins to freeze, the curve is here arrested and runs horizontal for a time. Heat here disappears in the changing of the body from a liquid to a solid, the internal temperature does not fall so long as any of the metal remains liquid, then after solidification the curve is regular again.

In the case of molten pure iron the cooling curve is the same as far as the freezing point (1,600 degrees C., 2,912 degrees F.), but the curve from there is not regular, there being two other arrests in the curve where it again is horizontal, at 890 degrees C. (1,634 degrees F.), and at 760 degrees C. (1,400 degrees F.). These two points are the critical points, Ar_3 and Ar_2, respectively, and by their existence it is proven that some internal change takes place by which sufficient heat is developed to make up for that lost in cooling. See Fig. 1.

A number of the elements can exist in different forms at the same temperature; for example, carbon exists as the diamond, graphite and charcoal. If the same weight of each

of these forms be burned the same weight of CO_2 is given off in each case, but the amount of heat developed is different. From this it is shown that the diamond, graphite and charcoal are the same element, but that their internal arrangement of atoms must be different with each; and it is most probably to this that these differences in characteristics are due.

As we speak of diamond-carbon, graphite-carbon and amorphous-carbon, just so we may speak of alpha, beta and gamma iron. The alpha modification is the one existing at

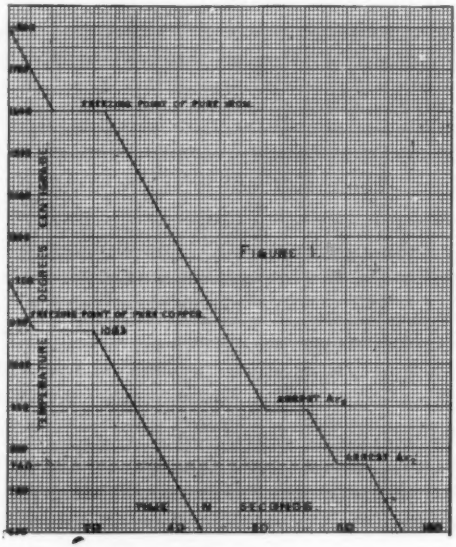

Fig. 1.

atmospheric temperature, and the iron here is soft and magnetic. Should this be heated, it still remains alpha iron until a temperature of 760 degrees C. (1,400 degrees F.) is reached, when a sudden change takes place. This change being evidenced by the absorption of heat and the loss of magnetic power; the iron here becomes beta iron, and, in addition to being non-magnetic, is hard. Upon still further heating, the iron at 890 degrees C. (1,634 degrees F.) again

changes its form and becomes soft and non-magnetic, and is then known as gamma iron.

The following summation is adopted:

Alpha iron, soft and magnetic, exists below 760 degrees C. (1,400 degrees F.).

Beta iron, hard and non-magnetic, exists between 760 degrees C. and 890 degrees C.

Gamma iron, soft and non-magnetic, exists above 890 degrees C. (1,634 degrees F.).

While the allotropic theory is not universally accepted, opinion is greatly in its favor; and it presents a simple explanation of certain observed facts. When, however, carbon is present in iron the allotropic changes are somewhat modified. With about 0.4 per cent. carbon the temperatures corresponding to Ar_3 and Ar_2 are lowered, and seem to run together at 700 degrees C. (1,292 degrees F.) with a pronounced evolution of heat. Also that when the temperature has fallen to 660 degrees C. (1,220 degrees F.) there is a very marked evolution of heat and the metal glows again with the added heat. This phenomenon has long been known as recalescence, although its cause was not properly understood.

In order to understand this peculiar phenomenon let us consider the cooling curve of a solder, consisting of two parts lead and one part tin. The melting points of lead and tin are respectively 326 degrees C. (619 degrees F.) and 231 degrees C. (448 degrees F.). About the melting point of lead the solder is fluid, and if allowed to cool it commences to solidify at about 240 degrees C. (464 degrees F.), but the temperature continues falling through a range of some 60 degrees C. (140 degrees F.) before complete solidification of the solder takes place. It thus assumes the condition of a pasty mass consisting of small particles of solid lead wetted by the molten alloy of lead and tin. When solidification begins to set in it is the pure lead that crystallizes out; this crystallizing out of the lead continues until an alloy of two parts lead and one of tin is left in the liquid state; this then solidifies at 180 degrees C. (356 degrees F.), and the whole

mass becomes solid. The alloy containing two parts lead and one part tin has the lowest melting point of any alloy of these two metals, and is called the "Eutectic Alloy."

The characteristics of a *chemical compound* are its absolute constancy of composition, concerning both the proportion and the nature of the elements it contains, and the complete merging of the properties of these elements into those of the compound.

If a body of any kind be dissolved in a liquid the liquid is called the *solvent* and the dissolved body is the *solute*. If the solute is evenly distributed throughout the solvent, and if the solution is homogeneous, the body is said to be of uniform concentration.

A *solution* and a *chemical* compound differ in that with a solution the proportions of the *solute* and the *solvent* may vary between wide ranges; *i.e.*, from a very minute proportion of the solute up to the saturation point. The temperature of the solution effects very much the saturation proportion, and this proportion, in the majority of cases, is lowered by a reduction in the temperature.

Liquids tend to freeze when they cool, and the freezing point of a liquid is in most cases lowered by dissolving a solid in it. For example, with common salt dissolved in water take a solution of one part salt and nine of water and subject it to the action of a freezing mixture. The temperature of the solution will fall below 0 degree C. (32 degrees F.), which is the freezing point of water, and the solution still remains entirely liquid until the temperature of —8 degrees C. (17.6 degrees F.) is reached, when the solid begins to separate. Remove the solid and examine it, it is found to be pure ice; the remaining solution thus becomes more briny, and a still further lowering of the freezing temperature is the result. The temperature falls and more ice is formed; the fall in temperature and the separation of more ice continue until the proportions in the brine are 23.5 per cent. salt and 76.5 per cent. water, and when the temperature falls to —22 degrees C. (—7.6 degrees F.) the

whole solution freezes completely. If, on the contrary, the solution contains more than 23.5 per cent. salt there will be no solidification until the temperature of —12 degrees C. (10.4 degrees F.) is reached, when the salt begins to separate out, and the separation of the salt with the lowering of the freezing point continues till —22 degrees C. (—7.6 degrees F.) is reached, at which point the composition of the brine is 23.5 per cent. salt and 76.5 per cent. water, and the solution then solidifies as a whole. It is thus seen that there is one particular ratio of mixing from which it is impossible to separate solute or solvent by lowering the temperature, for by so doing solidification takes place as a whole; this is the "*Eutectic Solution.*" The solute and the solvent seem to mutually reject each other at the moment of solidification. Should a solution contain an excess of solute or solvent over the eutectic proportion, and should the solution be cooled until it solidifies without removing the excess as it crystallizes out, the solid will consist of crystals of the excess constituent imbedded in the body of the eutectic.

Carbon is dissolved in molten iron in very nearly the same way that water dissolves salt, and as great a proportion as 7. per cent. of carbon will pass into solution under the proper conditions. The melting point of pure iron is about 1,600 degrees C. (2,912 degrees F.), but as the presence of salt in water lowers its freezing point, so does the presence of carbon in iron lower the melting point of the iron. Take a molten mass of iron containing upwards of 5 per cent. carbon, as the metal cools carbon is deposited in graphitic form, and when the temperature of 1,130 degrees C. (2,066 degrees F.) is reached the solution will contain 4.3 per cent. carbon. As this is the composition of the iron-carbon alloy of lowest melting point, *i.e.*, the eutectic alloy, and the whole mass becomes solid. The eutectic here is said to contain 2.3 per cent. graphitic carbon, while the other constituent is the solid solution of carbon in the gamma iron. The name "Austentite" is given to this constituent. When the molten alloy contains less than 4.3 per cent. carbon austentite, instead of

graphite, is deposited while cooling. As is the case with salt solutions, it is the excess water (ice) which crystallizes out first and the solution becomes more and more briny, so in the case of high carbon alloys, excess iron solidifies out of the solution first, most of the carbide remaining, and becomes more and more concentrated in the solution, until the temperature 1,130 degrees C. is reached, when the solution solidifies. The eutectic has not been traced in steels containing less than 1.2 per cent. carbon, so that at 1,130 degrees C. iron holds 1.2 per cent. carbon in solid solution. The concentration of austentite, like any other solution, may vary, so its carbon may vary between 0. and 2.2 per cent. The cooling curve is seen in Fig. 2, in the upper range, from 1,600 degrees to 1,130 degrees C. The carbon in the molten alloy is considered by some authorities as being in the elemental state and by others as being in the form of Fe_3C. (carbide of iron). Others claim that Fe_3C. above 1,050 degrees C. is broken up, that it becomes dissociated, and also that below 1,000 degrees C. iron can hold more than 1.2 per cent. carbon in solid solution due to its being in the carbide form. However, no matter which view is taken, the fact still remains that a number of important changes take place in the cooling solid. The allotropic modification of iron plays an important part with steel. "Beta" iron is capable of forming a solid solution with carbon, but "alpha" iron is not, because it does not dissolve *carbon*. The transformation range of "beta" to "alpha" iron lies between 890 degrees and 760 degrees C. in pure iron, but the presence of carbon retards the passage. There is always a certain amount of lag in the passage of an element from the upper to the lower, or the more stable, allotropic form. This lag is due to sluggishness on the part of the element to change its form, and with carbon this sluggishness is accentuated. This lag is seen by heating pure iron to the temperatures Ac_3 and Ac_2. These temperatures are a little higher (about 30 degrees C.) than Ar_3 and Ar_2, the ones on cooling. In a similar manner a solution may become super-saturated when carefully cooled

below its saturation point without any separation of solid. Just so a liquid may be cooled below its freezing point without freezing. Take for example water; it may be cooled below its freezing temperature without freezing, but if the smallest particle of ice be put in the super-cooled water, freezing immediately sets in and the temperature rises to its normal freezing temperature. So also may hyposulphite of soda be melted and cooled to the atmospheric temperature without solidification, but if the smallest crystal of salt be added crystallization will immediately set in, together with a rapid rise in temperature and the evolution of heat. Similar phenomena are supposed to exist in the cooling of cast iron and steel.

All steels freeze as solid solutions. Suppose steel with 0.5 per cent. carbon be considered. At a temperature of 1,600 degrees C. this will be indicated by point (b) in Fig. 2. This metal cools until the line AB is met, when freezing begins; it is for a while part liquid and part solid until the line Aa is met, it is then entirely solid and is a homogeneous solution of 0.5 per cent. carbon in iron. Consider also the point "c," Fig. 2, at 1,550 degrees C.; the constituents of the metal represented at this point are 99 per cent. iron and 1 per cent. carbon. This alloy cools, and when the line AB is crossed freezing commences and the cooling continues, the metal being part solid and part liquid until the line Aa is reached, when the solidification is complete and it becomes a homogeneous solution of 99 per cent. iron and 1 per cent. carbon. Other points representing various percentages of carbon may be taken and followed out on the diagram. In all these alloys, up to 2.2 per cent. carbon content, a solid solution of carbon in iron is the result; this solid solution is austentite. All steels are considered as being in the condition of austentite as soon as their solidification is complete. The diagram to the right of XY represents the freezing curve of cast iron. For example, take the point "e" at 1,400 degrees C. (2,552 degrees F.); the percentage of carbon is 2.5 per cent.; the mass cools, and when AB is reached, at about 1,135

degrees C., a large amount of the iron containing 2.2 per cent. carbon in solution will have formed as a solid, and the small amount of liquid solution remaining will be at the point " B,"

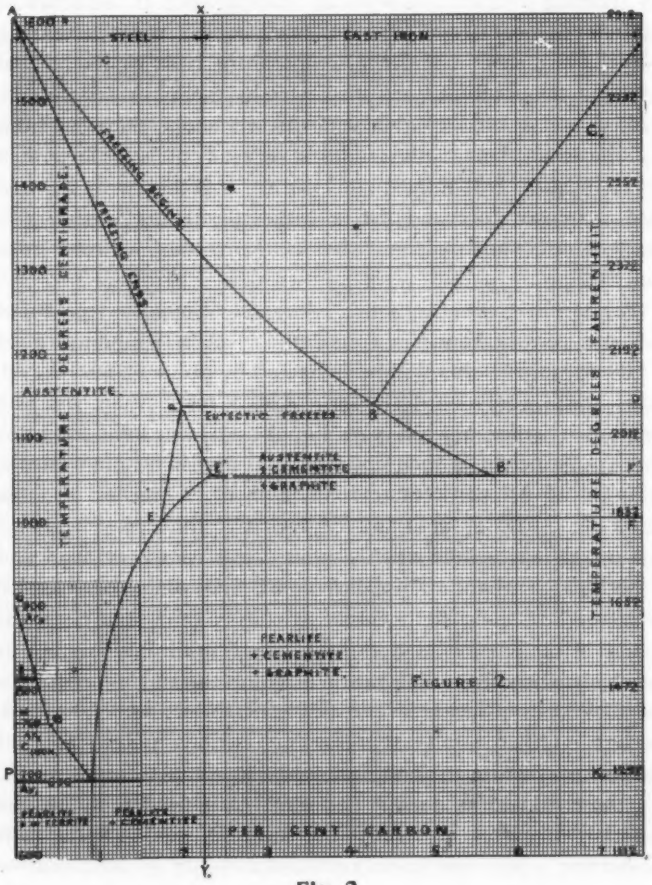

Fig. 2.

and upon further cooling the eutectic will cross the point B and its precipitation will there be complete. It there consists of the solid solution of iron, containing 2.2 per cent. carbon in flakelets, and crystals of carbon. That is to say, any

solution of iron containing between 2.2 per cent. and 4.3 per cent. carbon will, after freezing, consist of a solid solution of iron containing 2.2 per cent. carbon and a eutectic. As a summation, it may be stated that all solutions of carbon and iron containing less than 2.2 per cent. carbon will, upon freezing, consist of a solid solution of iron and carbon of the same chemical composition as the original solution. There will be no eutectic unless the carbon content be greater than 2.2 per cent. The effects of silicon and sulphur in iron-carbon alloys are most important in altering the diagram. Silicon makes the point "B" move to the left, *i.e.*, it causes the eutectic to have less than 4.3 per cent. C. Sulphur has the opposite effect, *i.e.*, to increase the total carbon which the eutectic may hold.

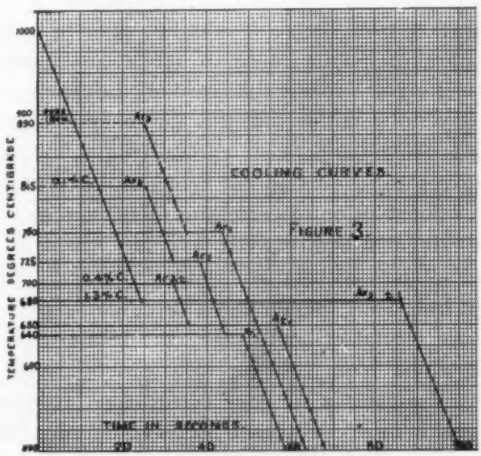

Fig. 3.

Fig. 3 shows the cooling curves of a few carbon steels obtained by Sir Roberts-Austen. The cooling curve of pure iron is also given. The ordinates represent temperature, and the abscissae intervals of time in seconds. The first arrest, Ar_3, at 890 degrees C. (1,634 degrees F.) indicates the passage in pure iron of gamma to beta iron. This tem-

perature is lowered to 815 degrees C. (1,499 degrees F.) by the presence of 0.1 per cent. of carbon in the metal. The second arrest, Ar_2, denotes the passage from beta to alpha iron and occurs at 760 degrees C. (1,400 degrees F.) with pure iron, but in metal containing 0.1 per cent. carbon this transformation takes place at 725 degrees C. (1,337 degrees F.). The arrest Ar_1 only occurs in iron containing carbon, and with a carbon content of 0.1 per cent. takes place at 640 degrees C. (1,184 degrees F.). The other curves shown may be readily followed, and it is seen that in steel with a carbon percentage of 0.4, Ar_3 and Ar_2 coincide at 700 degrees C. (1,292 degrees F.); and with 1.3 per cent. carbon Ar_3, Ar_2 and Ar_1 coalesce at 680 degrees C. (1,256 degrees F.).

The existence of Ar_1 is due to the formation of cementite or its separation from the solid solution. The heat developed by this separation is the cause of recalescence, and in the case of steel of 1.3 per cent. carbon the phenomenon is augmented by the heat developed by the other changes. Steel containing less than 0.9 per cent. carbon from 1,000 degrees C. (1,832 degrees F.) down to the temperature corresponding to Ar_3 is a solid solution of cementite (Fe_3C) in gamma iron, and in the critical range between Ar_3 and Ar_2 it is changed into a solid solution of beta iron with Fe_3C; the (Fe_3C) carbide in beta iron being known as "*martensite.*" Between Ar_2 and Ar_1 (the second critical range) the beta iron is decomposed into alpha iron. Alpha iron cannot form a solid solution, so the carbide of iron (cementite) separates, and with the ferrite forms pearlite. Steels containing more than 0.9 per cent. carbon when soft consist of pearlite and cementite; and those containing less than 0.9 per cent. carbon consist of ferrite and pearlite. The term, "Heat Treatment" is of comparatively recent origin, but, strictly speaking, it embraces processes that have been practiced for many years, viz: "tempering" and "annealing."

Tempering is the *partial* softening of hardened steel to remove some of the excessive brittleness of hardened steel and thus make it strong and elastic while it is still very hard.

Annealing is the *complete* softening of a piece of steel; *i.e.*, to say, as a rule, the obtaining of the greatest softness that is possible; or in any case to have the steel softer than tempering would make it. Annealing and tempering are often misused for each other.

The following test, known as the Metcalf test, is most interesting. Take a bar of steel, say about a foot long, and nick it at intervals of an inch. Place one end of it in a fire and heat to a white heat, the other end meanwhile being at a black heat. Remove the bar and cool it rapidly by quenching, then break the bar at the nicks; the crystals of the fracture at the heated end will be seen to be the largest, and they will grow smaller at each nick until at one of the nicks the fracture will show a close, velvety grain; this occurs at the point where the metal was at the temperature of restoration; and beyond the point of resoration the fracture will show the grain of the original bar before heating. This test may be used to compare steel suspected of being overheated with overheated steel of similar composition to determine the degree of overheating.

If a piece of steel be heated to a bright red heat and plunged into water, thus cooling the metal rapidly, it becomes very much harder, stronger and brittler. One requisite to make the steel harder is that the temperature to which it is heated be above the critical range. Take steel with a carbon content of 0.9 per cent., and if it be heated, no matter to any temperature, below "S" in Fig. 2, there is no increase in the hardness, even though the heated metal be cooled with extreme rapidity. If, however, the same steel be cooled from a temperature above "S," no matter how little above "S" the temperature may be, the metal will be hard and brittle. The quenching of a metal at a temperature slightly above the temperature corresponding to its recalescence point gives the maximum hardness that can be obtained with the metal. If the temperature at which quenching takes place be raised above "S," with steel of 0.9 per cent. carbon the *grain size* is increased; if the carbon be less than 0.9 per cent.

the best temperature for hardening is just above the line "GOS," that temperature giving the maximum hardness with the best grain size. For steel containing more than 0.9 per cent. carbon the best temperature for hardening is just above the line SK, and that temperature gives the best *grain size*, but by cooling along Sa *greater hardness* may be obtained.

Every increase of carbon in steel gives increased hardness to the metal; not only is this true with steel in its natural state, but the influence of carbon is even more important when steel is subjected to heat-treatment. With a steel its hardness increases with the rapidity with which the cooling above the critical temperature takes place. Hence the explanation for the results obtained when three pieces of steel of the same composition are heated to the same point above the critical temperature and cooled in mercury, iced water and oil respectively. The piece cooled in mercury will be the hardest, the one in iced water the next hard and the one cooled in oil the least hard. The main requirement with steel in order to produce hardness is that it be heated above the line "PSK," *i.e.*, to a temperature where at least some of the solid solution of iron and carbon is formed.

Both the "carbon" and the "allotropic" theories agree that at temperatures above the critical range the molecules of the steel are in a hardened condition. Cooling from this point and across the critical range the molecules change slowly from the hard to a soft state; if the metal is allowed to cool slowly it will be soft, but the sudden cooling arrests the metal's slow cooling and stops the molecules in their hard state. The question at variance between the two theories is, "what is it that makes the molecules hard when the steel is above the critical temperature?"

It is hard to imagine how the molecules can be in a hard state while the metal is molten, but this may easily be understood when one remembers that a ball of wet clay is soft and plastic while the grains of sand composing it may individually be very hard, in fact hard enough to scratch glass. The

carbon theory assumes that the hardness of steel is due to the carbon dissolved in it. This theory has in its favor the fact that hardness varies almost directly with the carbon content. On the contrary, it is advanced that the amount of carbon is too small to produce the great hardness encountered in steel, and that the presence of carbon in metals does not induce hardness in any but iron. The great argument in favor of the allotropic theory is that tests have been conducted in laboratories with iron practically free from carbon and that the beta and gamma modifications of iron were found to be harder than the alpha form. Another point in favor of the allotropic theory is that steel, upon cooling slowly through the critical range, loses hardness slightly before the carbon separates out of the solution. Then, too, the greater hardness of high-carbon steel over low-carbon metal is attributed to the fact that the presence of carbon retards the change from gamma to beta and to alpha iron, and, on account of this change being slower with a high carbon content, more of the iron upon rapid cooling is retained in the gamma and beta forms.

To sum up: The hardening of steel is due to the preservation of the austentite condition by cooling the metal quickly so that not sufficient time is given for the austentite to change into pearlite, etc., and once the metal is cooled, the metal is locked in the unstable condition of austentite and cannot then change on account of the rigidity of its molecules which this low temperature causes.

In tempering a piece of hardened steel we heat it say to 220 degrees or 320 degrees C., and this slight heating lessens the existing strains a bit, and thus causes a reduction in the molecular rigidity of the austentite, and the unstable austentite has a tendency to form pearlite. The higher the degree of heating the greater is the tendency for the austentite to break up, on account of the molecular rigidity being further lessened. Should the piece of hardened steel be heated to a low red heat and allowed to cool slowly there is sufficient

time for the austentite to change to pearlite, etc.; the metal is then said to be annealed.

Should a piece of steel be heated to a high temperature, say 1,100 degrees C. (2,012 degrees F.), and then cooled (either slowly or rapidly) *without subjecting it to strain* it will be coarse grained. Even the best quality of steel may be rendered coarse grained and practically unfit for use by overheating it.

Should a piece of steel be heated a number of times above "PSK" (Fig. 2) and cooled again it is only the last heating that leaves its impression upon the structure of the metal, because each crossing of the line on the way up removes the effect of the previous treatment. This is only true provided that the previous heating temperature was not near the melting point, *i.e.*, to say when the metal has become what is technically known as "burnt." Burning sets in when the temperature crosses the line "Aa" (Fig. 2). As a safety precaution with burnt steel it is advisable to scrap the metal for remelting.

In hardening steel it should be heated just above the recalescence point, as heating beyond that is apt to produce open grain. Should a piece of steel be overheated but not burned it may be restored by the following operation: heat the metal to the Ac_2—$_3$ point, *i.e.*, the point where the metal becomes non-magnetic. This is accomplished practically by heating the steel and from time to time removing it and placing it near a magnetic needle, to determine when it has lost its magnetic quality. Should the steel be a large piece, a small piece of the same metal is heated alongside the large piece and used for testing for magnetism. Steel is said to be hardened when the metal is quenched in water or some other medium which takes away the heat as fast as or faster than water. Steel partially hardened is said to be tempered, and steel with all of its hardness removed is said to be annealed.

From what has been said it might be wondered why steel is heated to 1,100 degrees C. to 1,350 degrees C. (2,012 degrees F. to 2,462 degrees F.) for forging or rolling. Were

there no strains impressed on the metal the crystals would be large, but the mechanical pressure upon the metal breaks up the crystals and reduces them to small size. The resulting size of the crystals is dependent upon the finishing temperature of the forging or rolling. So steel finished at 1,100 degrees C. (2,012 degrees F.) has a coarser grain than the same steel finished at 900 degrees C. (1,652 degrees F.).

William Campbell has found that the best finishing temperature for 0.5 per cent. carbon steel is just where the ferrite begins to separate from the solid solution. This occurs just below "GOS" (Fig. 2). Work below that temperature greatly increases the brittleness of the metal, while finishing work above that temperature gives lower strength. Tentatively, it may be said that the best results may be expected with steels if the mechanical work upon them is completed at a temperature which brings it exactly upon the line "GOS" or "SK."

In heating a tool for tempering or forging great care should be exercised that the metal be heated uniformly. Tools with sharp edges and corners should be allowed to cool a little before quenching, as the edges are generally hotter than the main mass of the piece when it is removed from the heating chamber. Very soon the temperature is equalized and the tool is ready for quenching. Large tools are more difficult to harden than small ones and should be raised to a slightly higher temperature, on account of the more rapid quenching of the smaller article. With large tools the time for heating them is longer, consequently oxidation is apt to take place unless their surfaces are coated with some protecting material. But the coating must be some substance that will not affect the steel. Common salt, borax or soap may be used; they are sprinkled over the tool and a glaze is formed on the metal. Heating for tempering is sometimes done by immersing the article in a bath of molten salt, nitre or lead, or heated oil. Heating a tool in a blacksmith's fire or a furnace filled with flame subjects the metal to the fuel and the products of combustion, thus making it possible for the article to be injured by the absorption of

sulphur. A very good way to heat for tempering is to place the metal in a muffle furnace where the flame does not come in contact with the article. In this type of furnace there is more liability to oxidation, but protective coatings may be used to prevent this.

Stead's Brittleness.—This is a term applied to a brittleness which was discovered by Stead in ferrous metals having a carbon content of from 0.025 per cent. to 0.12 per cent. When they are subjected to prolonged heating between 600 degrees C. (1,112 degrees F.) and 750 degrees C. (1,382 degrees F.) they become coarsely crystalline and brittle. Upon reheating to 900 degrees C. (1,652 degrees F.) the coarse crystalline structure disappears and the metal is refined.

The following authorities were consulted in the preparation of this paper: W. J. Keep, H. H. Campbell, J. H. Stansbie, Bradley Stoughton, H. M. Howe, T. Turner, R. H. Thurston, William Metcalf, and the "Journal of the Iron and Steel Institute," 1909 and 1910.

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RESULTS OF TRIALS OF THE ANTI-ROLLING TANKS AT SEA.

By Herr H. Frahm.

Read at the Spring Meetings of the Fifty-second Session of the Institution of Naval Architects, April 7, 1911.

GENERAL OBSERVATIONS ON ANTI-ROLLING TANKS.

Anti-rolling tanks have been developed by the author in the shipbuilding yard of Messrs. Blohm & Voss, Hamburg, during the last few years. In explanation of the action of the anti-rolling device, some general remarks on their nature will be useful.

The idea of employing free water, placed in the hull, for damping the rolling motions is not new. In the old iron-clad *Inflexible* a parallel-sided athwartship chamber, partly filled with water, was intended for this purpose. Sir Philip Watts in 1883–5 read before this Institution papers describing very thoroughly the experiments and results obtained with these water chambers.

The experiments with the *Inflexible,* which proved the efficiency of these tanks, took place in the Mediterranean off Alexandria, but to our knowledge the trials have not been repeated; apparently it was impossible to master the water, rushing freely from side to side in the tank. Besides, the latter, which extended right across the ship, occupying a full deck's height, required too much room, which could not be spared from other purposes. After these trials water chambers do not seem to have been made use of for any length of time.

The present device avoids the disadvantage of having such a big tank, while it is at the same time far more efficient against rolling. This result has been obtained by the most careful

application of the laws of resonance, the great importance of which in engineering generally is recognized more and more every day.

Under the effects of resonance, bodies that can oscillate about a condition of equilibrium are made to swing severely under comparatively small impulses, as soon as the period of oscillation of the impulse is synchronous with the individual periods of the respective bodies. The phases of oscillation of the body and of the impulse are deferred by 90 degrees.*

A ship constitutes a body of this kind, as she will oscillate under the impulse of the waves. As is well known, a ship will roll almost exactly in the periods of her individual oscillations. This has been proved by practical observations at sea, and is confirmed by the diagrams given in Plate II, which were recorded on S.S. *Ypiranga* and *Corcovado*, as well as by experiments on models (particulars of which are given in the Appendix). It can thus be shown that a ship will roll in her individual number of oscillations, even if the impulse be more or less irregular.

Large heeling amplitudes can only be produced on a ship, if she is struck by a series of comparatively regular waves, and struck in the measure of her individual number of oscillations. Under such circumstances the influence of resonance will be promptly felt, and there will be an increase of the angle of heeling from wave to wave. All large rolling amplitudes observed in practical seafaring are due merely to the influence of resonance between wave and vessel.

On this fact the present device has been based. It utilizes a secondary and artificial resonance in order to annihilate the influence of the primary resonance between waves and ship.

This secondary resonance is introduced by means of a U-shaped tank, located athwartship and extending from side to side, in which a water column can oscillate with the same number of swings per minute that are peculiar to the ship herself. The tank is designed in the form of a communicating tube (see

* This law can be demonstrated with a special apparatus based on the well-known double pendulum.

Fig. 8, Plate III), and consists of two vertical parts S at both sides, and a connecting part H. The water fills the latter entirely, and about one-half of the vertical tank.

The individual number of oscillations of the water column thus enclosed will, of course, depend upon the dimensions of the whole tank, and it will, therefore, be the designer's task to calculate them accordingly.*

According to the law of resonance, the phases of the impulse of waves and ship are deferred by 90 degrees, that is to say, the ship lags a quarter of her full period behind the wave, or, in other words, the ship arrives at the heeling climax a quarter of a period later than the wave in its advancing movement arrives at its maximum slope towards the ship. The same law naturally applies to the rolling of the ship, and the oscillating movements of the tank water produced by the former. Also, in this instance, the phases are deferred by 90 degrees, and the tank water will reach its highest or lowest level in the vertical parts a quarter of a period later than the greatest heeling of the ship to one side or the other.

Consequently there results a total difference between the impulse of the waves and the oscillations of the tank water of $90° + 90° = 180°$ and the latter will act just in the opposite direction to the impulse of the waves. Thus the ship will heel only as far as the water, under the action of secondary resonance, rises or falls in the vertical tanks to such an extent that the heeling moment imparted to the ship by the waves is balanced by the opposite turning moment, produced by the oscillations of the water. The increment of heeling, from impulse to impulse, in a free ship, cannot take place now, and the rolling motions will be limited to such as will be sufficient to produce the necessary oscillations of the tank water in accordance with the foregoing explanation.

It may here be pointed out that Sir Philip Watts had already found out that the damping efficiency of his water chambers was at its best at a certain depth of water, *i.e.*, a depth suf-

* The effect of this device, and principally the influence of the secondary resonance, can be explained with this experimental apparatus by simply adding a third pendulum.

ficient for producing a wave moving from side to side in the same time as the ship performs half a heeling motion.

In his device the principle of resonance also existed, but this fact was not discerned or stated at that time, this being reserved to the present invention. In the earlier device a free wave was breaking from side to side in an open chamber, while in the present instance a homogeneous water column is enclosed in a well-dimensioned tank, where it performs steady and uniform movements, which at any moment can be controlled by blocking the connections.

A great advantage of this invention is that only a comparatively small connecting passage is required, whereas the earlier water chambers wholly occupied the space from side to side to the full deck's height.

Professor Biles has improved upon the open water chamber (especially for ships of slow period) by narrowing the bulkheads of the chamber, thus leaving the tank somewhat narrower in the center of the ship. But the free-water surface remained, and consequently it was impossible to keep the water under control, nor was it possible to arrive at the efficiency which is obtained with the present arrangement.

The author has to thank Sir William White and Sir Philip Watts for informing him that the first proposal for the *Inflexible* was somewhat different. It was intended to arrange a cross connection between the cofferdams, out of which the cork filling was to be withdrawn and replaced by water. The cork, stored in other parts of the ship, was only to be replaced on the declaration of war, but this proposal was not agreed to, as it was considered that the replacing of the cork would take too much time, and the water chamber, as already described, was adopted instead.

Although this proposal was somewhat similar in appearance to the present device, it must be stated that the fundamental reason of the great efficiency of the new anti-rolling tank, viz: the effect of resonance between the enclosed water column and the ship's oscillations, and the phase difference of about 90 degrees, were not discerned at that time, and, consequently, could not be made use of.

The connection (L) between the upper parts of the vertical tanks is provided with a throttling apparatus (D) which is of great importance to the present invention. It serves first of all to stop the movement of the water by simply blocking the connection. Besides, the air passing through the upper connection, in accordance with the rising or falling of the water in the vertical tanks, can be more or less throttled in order to adjust the oscillations of the water to the condition of the sea.

EXPERIMENTS ON S.S. "YPIRANGA" AND "CORCOVADO."

The first experiments to be referred to took place on the passenger steamers *Ypiranga* and *Corcovado*. These ships, although new and of excellent construction and fitted with bilge keels 1 foot in depth over two-thirds of their length, were said to be bad rollers, and, therefore, made excellent subjects for trial.

The principal data of both vessels, which have been built as sister ships, are as follows:

Length between perpendiculars, feet.....	447
Extreme breadth, feet.....	55
Loaded draught, feet and inches.....	25-5
Loaded displacement, tons.....	14,100
Speed, knots.....	13

The arrangement of the tanks is shown in Figs. 1, 2, 3 (Plate I). There are two sets of anti-rolling tanks, one fore and one aft, placed at the position of the masts between the existing loading hatches on the uppermost deck. Each set consists of two tanks, of 9 feet height and 10 feet athwartship breadth, placed close to the shell plating, with a lower connection of 1 foot 6 inches height, carried over the deck (see Fig. 2). The length of each basin in the ship's longitudinal axis is 25 feet. The upper parts of the side tanks are connected by a flying bridge, in order to get the necessary space for the location of the winches, which hitherto had been placed on the continuous deck. Below this bridge are placed the air

connections of the tanks, provided with valves for throttling or entirely cutting off the connections, which will regulate or stop the motion of the tank water.

For better guidance of the water, rushing in and out of the tanks in a seaway, special guide plates have been provided. The tanks are filled by the wash-deck pump, and discharge directly outside through a foot valve. In the middle of the lower connection a water gage is arranged. The total quantity of water in each tank is 94 tons.

The first experiments were made at the end of March, 1910, during the first outward trip of the *Ypiranga,* after she had been fitted with the anti-rolling tanks. During this voyage, and in the presence of quite a number of prominent nautical and technical experts who went out in her to Lisbon, the damping effect of the anti-rolling tanks, although the sea was not very rough, was proved in the Gulf of Biscay and off the north coast of Spain. It was therefore resolved to have her sister ship, *Corcovado,* also fitted with anti-rolling tanks.

During the experiments the rolling motions of the ship with the tank partly out and partly in action were recorded by means of a specially arranged recording pendulum.

In the course of last summer the experiments were continued on the *Ypiranga,* as well as on the *Corcovado,* during their regular voyages between Hamburg and Buenos Aires or Mexico, under many different conditions of sea; some characteristic diagrams are given in Fig. 4 (Plate II) for the *Ypiranga,* and Fig. 5 for the *Corcovado,* and the excellent damping efficiency of the anti-rolling tanks was confirmed in every way. In special cases heeling angles of 11 degrees to either side were reduced through the action of the anti-rolling tanks to only 2 to 2½ degrees. It was stated that the influence of the anti-rolling tanks on the health of the passengers was very favorable, and that these ships, which had previously been unsteady and much given to rolling, had become, through the addition of this device, quiet, comfortable vessels.

Though both ships had occasionally met with bad seas on their first voyage, the anti-rolling device had still to be experi-

TABLE I.

Date.		Draught.	Displacement.	1. Without water in the anti-rolling tank, and without taking into consideration the free water in the partly filled compartments of the double bottom.						2. Without water in the anti-rolling tanks, the partly filled compartments of the double bottom taken into consideration.				3. Anti-rolling tanks filled and partly filled compartments of the double bottom taken into consideration.			
				MK	GK	MG	R	t	s	MG	R	t	s	MG	R	t	s
1910		*feet.*	*tons.*	*feet.*	*feet.*	*feet.*	*feet.*	*sec.*		*feet.*	*feet.*	*sec.*		*feet.*	*feet.*	*sec.*	
Dec. 7	Dept. Havana.....	22.9	12,000	23.35	19.54	3.81	21.55	6.10	4.92	3.55	21.55	6.32	4.75	3.22	21.55	6.64	4.52
Dec. 12	Experiment	22.4	11,600	23.40	19.79	3.61	21.55	6.27	4.79	3.35	21.55	6.50	4.62	3.02	21.55	6.85	4.38
Dec. 22	Arr. Vigo	21.15	10,730	23.60	20.52	3.08	21.30	6.70	4.48	2.82	21.30	7.01	4.28	2.49	21.30	7.45	4.03

TABLE II.

Date.		Draught.	Displacement.	1. Without water in anti-rolling tank, and without taking into consideration the free water in the partly filled compartments of double bottom.						2. Without water in anti-rolling tank, and partly filled compartments of double bottom taken into consideration.				3. With anti-rolling tank filled, and partly filled compartments of double bottom taken into consideration.			
				MK	GK	MG	R	t	s	MG	R	t	s	MG	R	t	s
1910		*feet.*	*tons.*	*feet*	*feet.*	*feet.*	*feet.*	*sec.*		*feet.*	*feet.*	*sec.*		*feet.*	*feet.*	*sec.*	
Dec. 3	Dep. Hamburg..	21.5	10,900	23.6	19.63	3.97	21.90	6.07	4.95	3.78	21.90	6.25	4.81	3.45	21.95	6.52	4.61
Dec. 7	Experiment	20.9	10,540	23.7	19.82	3.88	22.05	6.20	4.85	3.68	22.0	6.35	4.73	3.35	22.05	6.65	4.52
1911																	
Jan. 1	Arr. Mexico.....	17.7	8,670	24.6	22.01	2.59	22.40	7.67	3.91	2.46	22.3	7.87	3.82	2.13	22.40	8.46	3.55
Jan. 4	Dep. Mexico.....	21.9	11,220	23.5	19.62	3.88	21.90	6.14	4.89	3.61	21.9	6.36	4.72	3.28	21.90	6.67	4.50
Jan. 20	Experiment	19.9	10,000	23.9	20.48	3.42	22.30	6.65	4.51	3.15	22.2	6.82	4.40	2.82	22.30	7.32	4.10
Feb. 4	Arr. Hamburg...	18.0	8,870	24.5	21.65	2.85	22.40	7.32	4.10	2.69	22.4	7.54	3.98	2.36	22.40	8.05	3.73

mented with during really heavy weather. Ample occasion for this was afforded last winter during the heavy gales in the Atlantic. The *Corcovado* on her tenth trip met with very rough weather in the middle of the Atlantic. The diagrams of December 12, 1910, are shown in Fig. 7 (Plate II).

The curves, which were continuously recorded from 11:30 A. M. to 5:55 P. M., and which are partly reproduced in Fig. 7, show, for the undamped ship, rolling angles ranging up to 18 degrees on each side of the vertical. At noon, first the fore tank was set free, and immediately a regular decrease of the rolling was experienced. The aft tank, hitherto empty, was filled up during the next few hours, and put into action at 3:52 P. M., and from that very moment the ship showed an extraordinary regularity in her movements. The greatest heeling angles were scarcely more than 3 degrees to 4 degrees, and further diagrams, the recording of which was continued until the next port, showed exactly the same result.

From the diagrams it is obvious that, under the then existing conditions of sea and weather, one tank by itself had a favorable influence on the ship, but could not entirely damp her rolling.

For immediate comparison it ought to have been possible to suddenly block both tanks at the same time, in order to obtain diagrams of the unquenched rolling under the same conditions; but during the heavy weather this could not be allowed by the captain on account of the passengers.

The general conditions of stability of the *Corcovado* during this voyage, and especially during the experiments just mentioned, are shown in Table I.

The value of the different letters in Tables I and II is as follows:

M=Transverse metacenter.
G=Center of gravity.
K=Upper edge of keel.
R=Radius of gyration.
t=Duration of a single period, in seconds.
z=Double oscillations per minute.

The *Ypiranga* on her outward trip to Mexico (December 7, 1910) also met with exceedingly heavy seas and a strong gale in the Bay of Biscay. A diagram taken under these conditions from 11:20 P. M. to 12:25 A. M. is shown in Fig. 6 (Plate II), but only the curves with both tanks in action have been recorded; the average heeling angles range from 4 degrees to 5 degrees, and only occasionally somewhat wider ranges are to be found. In this case also, consideration for the passengers' comfort and the ship's safety did not admit of blocking the tanks for immediate comparison. Thus the undamped motion of the ship could not be recorded, but it was supposed by the leading men on board that without the damping action of the anti-rolling tanks she would have rolled to more than 25 degrees on either side.

During the homeward trip of the *Ypiranga,* and under slightly easier conditions in the middle of the Atlantic, experiments were made, on January 20, 1911, with the aft tank alone. The corresponding diagram recorded from 4:35 to 5:40 P. M. is shown in Fig. 6, second row.

From 4:50 to 5:24 P. M. the tank was blocked, with the result that during this time the heeling angles increased to a maximum of 18 degrees, whereas with only one tank in action a maximum of but 7 degrees was recorded. The second tank had, unfortunately, to undergo some small repairs and could not therefore be used to complete the experiments.

The conditions of stability of the *Ypiranga,* both for outward and homeward voyage, are shown in Table II.

The data obtained during the experiments lead to the general conclusion that the anti-rolling tanks added on the S.S. *Ypiranga* and *Corcovado* have fully done their duty under very different conditions of weather and sea, as well as of loading. Under the average and more favorable conditions, which are usual, the efficiency of the anti-rolling tanks is absolute, inasmuch as all rolling motions are almost entirely damped. But they have also undoubtedly a very favorable influence under the worst conditions. It is a fact that on both ships, since the anti-rolling tanks have been added,

safety boards, which had always to be used before, are now no longer necessary, even with the weather and sea at their worst. Moreover, the passengers, during the frequent experiments with the tanks out of action, urged to have them put on again, which speaks strongly in their favor.

When in full action these tanks exert a turning moment of 2,790 foot-tons, thus counteracting wave impulses of equal turning moment. This figure is of great importance for judging the whole problem, as it throws some light on the power required for effectually damping the rolling of a ship like those mentioned. In order to obtain equal efficiency, the same power ought also to be exerted by any other anti-rolling device, *e.g.*, the gyroscope, but it is doubtful if it will be possible to develop the latter sufficiently.

Experiments with anti-rolling tanks have been also made on other ships, details of which may be seen in last year's Transactions of the *Schiffbautechnische Gesellschaft*, which include a paper on this subject read by the author. Further experiments are to be made, as quite a number of shipowners (Hamburg-American Line, Hamburg-South American Line, German East African Line, Woermann Line) are having their new passenger ships fitted with these tanks.*

The first of these vessels, the S.S. *General* (13,620 tons loaded displacement), of the German East African Line, has now been completed; she started on her maiden trip to East Africa at the beginning of March. She is fitted with a tank as shown in Fig. 10 (Plate III), and has met with a storm during which the anti-rolling device proved highly satisfactory.†

As a rule all ships fitted with anti-rolling tanks are also

* In England also this question is being followed with the greatest interest, and negotiations are proceeding with some of the largest firms with a view to fitting such tanks in vessels under construction, and also in existing ones.

† According to detailed reports the S. S. *General* met with strong winds and a heavy sea in the Bay of Biscay on March 8 and 9, 1911, which made her roll 14 degrees on either side, when the tanks were out of action. The following experiments were then made: First, the smaller (fore) tank was put on, with the result that the heeling angles were at once reduced by 7 degrees or 8 degrees. Then the larger (aft) tank was also put in action, and the rolling was reduced to 3 degrees in either direction. By blocking the latter tank the rolling amplitudes rose again to 5 degrees, and, with both tanks out of action, 14 degrees. The diagrams recorded during these experiments are not yet available for publication.

fitted with a recording pendulum, in order to collect data, which will furnish additional records of interest.

ON THE LOCATION OF ANTI-ROLLING TANKS.

The practical location of the tanks is determined by various considerations. As they only exert a turning moment round the longitudinal axis, theoretically it will be of no importance whether they are placed amidships, fore or aft. Practically the midship position will always be preferred, because this is the broadest part of the vessel, and, besides, the yawing of the ship, *i.e.*, the lateral sheering under the impetus of cross seas, is less felt here.

These sheering motions may possibly diminish the efficiency of the tanks by accelerating the movement of the water contained in the cross connection; but this is only likely to happen in stiff vessels with a high period of frequency, which consequently require the cross communication to be rather wide. On normal vessels, *i.e.*, not too stiff, the tanks may as well be placed fore and aft. Instead of only one big tank, two or more smaller ones may be employed and placed in different parts of the ship.

As to the height above load water-line, no general rules can be established. The best efficiency will be obtained by placing the cross connection above the center of gravity. In this case the hydrodynamical action of the water in the connection will increase the statical action of the side tanks; on the other hand, it would be quite possible to place the connection below that center, if the tanks can be enlarged accordingly. This increase in size will be small in ships with slow periods, but more important in stiff vessels with comparatively short periods.

It will now be shown by some examples how the tanks, the dimensions of which should be calculated according to the rules given in the author's paper published in the Journal of the *Shiffbautechnische Gesellschaft* (1911), are to be located in the ship. In Fig 9 (Plate III) are shown the section and

part of the profile of a very large passenger and cargo steamer of 51,000 tons in which the tank is located in the middle of the ship, somewhat above the load water-line, and occupies a space hardly available for other puropses. The wing tanks are 16 feet 5 inches broad, and are carried through two decks. The cross connection is 2 feet high, and placed below the lower deck. Special guide plates are arranged to check the flow of water rushing in and out. The cross connection has also been furnished with vertical guide plates, in order to proportion the section of the water to the values obtained by calculation. It will be noted that the size of the tanks, although designed for the high damping characteristic of 3.2 degrees,* is of practically no importance compared to the size of the ship.

In Fig. 10 (Plate III) the arrangement of the anti-rolling tank on board the German East African liner *General* is shown. The feeding and discharging arrangements, as well as the air-throttling valves, are operated from the fore stoke hold.

Figs. 11 and 12 (Plate III) show the arrangements for an anti-rolling tank on a modern 25,600-ton battleship which has been designed for a foreign government. The design of the tank is based on a damping characteristic of 2 degrees. From Fig. 11, showing a cross-section of the ship, it will be seen that the tank follows the double bottom, the latter being carried up to the armor deck. The cross connection is 1 foot 10 inches high, and graded into the wing tanks, the latter being each 6 feet 5 inches wide. Between the inner skin of the side tanks and the longitudinal torpedo bulkhead an accessible passage of 2 feet 7 inches width is left. The air connection is located between the deck beams below the armor deck. The throttling apparatus, in the form of a valve, can be actuated from the center-line passage.

Where great stresses are to be expected, such as when docking the ship, the cross connection has been stiffened by special cast-steel pieces. The frictional resistance of these stiffeners, as shown by experiments made for the purpose, is so insignificant that these additions in no way prejudice the action of the

* A definition of the "damping characteristic" is given in the author's paper referred to above.

tank. The latter, in its longitudinal direction, has been divided in four compartments, each fitted with separate feeding and discharge valves and separate air connections.

The total length of the tanks is 83½ feet, which appears rather large, but it has to be borne in mind that in this instance the wing tanks could be given a breadth of only 6 feet 5 inches.

RECORDING PENDULUM FOR ROLLING MOTIONS.

For the practical experiments a reliable recording instrument could not be dispensed with. For ordinary ship's use, therefore, a simple apparatus has been designed, based on the pendulum and constructed from the fundamental ideas first propounded by Froude. This apparatus has been used primarily for the experiments already described. For the sake of brevity we may call it the "roll pendulum" (Fig. 13, Plate III). Its essential part is a pendulum, placed on knife-edge bearings, of the long swinging period of 63 seconds, corresponding to 0.96 double oscillations per minute. The pendulum is formed by a beam with identical weights at both ends, the center of gravity being placed about 1/32-in. below the knife edge *c* by means of a small additional weight *d*.

The knife edges of the pendulum are placed in the bearings of a frame connected to the ship. The turning movements of ship and frame in relation to the pendulum are recorded by pencil *g* on drum *e*, the latter being driven by electromotor *h*. The pendulum, on account of its great inertia and the very small distance of its center of gravity from the axis of rotation, will keep its vertical position without following the rolling motion of the ship, as long as the frequency of oscillation of the ship is at least twice that of the pendulum.

As the rolling motion of ships will always keep above 2½ oscillations per minute, the diagrams recorded by the apparatus can be regarded as reliable, and free from error, a fact that has been proved by experiments carried out in the Imperial Navy.

It should, however, be mentioned that the pendulum, under certain circumstances, may be influenced by lateral acceleration. The action of rudder will cause a turning motion of the ship round a vertical axis, which may lead to a movement of the pendulums. Such motions, if they occur, will be noticed in the diagram as long curves, intersected by the rolling curves of the vessel, and will be most apparent if the rolling amplitudes are comparatively small.

The rolling amplitudes in such cases have to be counted from the undulating line, and the diagram has to be corrected by stretching the curves of the easy motions of the roll pendulum and plotting the rolling amplitudes from the stretched line.

FINAL DEDUCTIONS.

The foregoing explanations may be summed up as follows:

The present arrangement offers for the first time the possibility of building ships of any size and stiffness so that their steadiness against rolling can be ensured. This is of special importance, as it will no longer be necessary to design vessels, especially passenger boats, as tender as possible in order to make them "quiet and steady." The anti-rolling tanks will allow greater freedom when settling the beam of ships, which is of fundamental importance for their stability. Nowadays, with the increasing tendency towards large ships, this point is of the greatest importance, as only length and breadth can be augmented, but not the draught, which, on account of the limited depth in harbors and rivers, hardly admits of any increase at all. Also, for other reasons, greater beam will be very favorable for some types of passenger boats, because they will thus be able to carry a larger number of passengers. This will raise their earning capacity, and admit at the same time of a more favorable arrangement of saloons and promenade decks.

The quantity of water required for the tanks will, of course, increase with the ship's beam. It will, therefore, be a question of careful study and calculation to ascertain to what extent

the breadth may be increased without arriving at an excessive size of the anti-rolling tanks. It has, however, been shown by calculation that it will be quite possible to build passenger boats with the usual high superstructure up to 120 to 130 feet beam and a draught of 34 to 36 feet, which, with the aid of the anti-rolling device, will be perfectly steady, the necessary water required being only 1.3 to 1.5 per cent. of the displacement.

Further advantages of the damping action of the tanks consist in the smaller loss of speed in unfavorable seas, the decrease of the transverse stresses in the hull, and improved steering and maneuvering qualities of ships.

The tanks can also be easily built without great additional cost into existing ships, thus essentially improving their seagoing qualities.

The weight of the tank itself is small, compared with the total weight of the ship. As to the tank water, it may be mentioned that it will only have to be carried if rolling is going on or is to be expected. The tanks can also be worked with fresh water, carried for ship purposes, or liquid fuel, if on board.

Hitherto the application of the anti-rolling tanks for merchant vessels has principally been dealt with, but they will doubtless prove of the greatest importance also to men-of-war. A steady ship will always be welcome to passengers, but a steady gun platform is almost indispensable to the gunner. Future naval battles will depend to an extraordinary extent on the guns, and every device tending to improve the efficiency of ordnance may be regarded as a valuable improvement in an up-to-date battleship.

Rolling in a man-of-war is not only a serious drawback for her gunnery, but it may also raise her armor belt above the water surface, thus exposing her vital parts to the enemy. For the location of the tanks on board warships some technical difficulties may arise; but, in view of the importance of this question, it seems likely that a compromise will be arrived at in this instance, as in other questions of naval architecture.

Finally, it is worth mentioning that the author has worked out still another form of anti-rolling device, which offers great advantages, especially for use on board men-of-war, because it dispenses with the cross connection. In this arrangement the wing tanks, through specially shaped and dimensioned holes in the shell-plating, have direct communication to the sea. With this form of tank, which in model experiments has given good results, further experiments on a larger scale are to be made shortly, and the author will be pleased to publish the results later on.

APPENDIX I.

MODEL EXPERIMENTS.

Anti-rolling tanks have been carefully studied on models, in order to obtain the necessary fundamental data. For this purpose models of different types of ships were made, with all appendages, such as rudder, bilge keels, propeller bosses, &c., and the static and dynamic conditions of real ships were reproduced with the utmost care. Assuming a linear reduction of $\frac{1}{a}$ for the model, the proportions for the corresponding values of the real ship are as follows:

Weight,	$1 : a^3$
Metacentric height,	$1 : a$
Moment of inertia, referred to the longitudinal axis,	$1 : a^5$
Number of oscillations,	$1/\sqrt{a} : 1$

The principal object of the experiments was to obtain some data for the estimation of the eventual behavior of the respective ships in a seaway, but the water surface of the experimental tank proved too small for producing waves of the necessary regularity, size and shape. Consequently a substitute had to be devised. To this end a plate spring was used, moved to and fro by a crank gear and connected to the model.

Thus, oscillating impulses were imparted to the model, of much the same kind as those of waves of a certain slope passing underneath a ship. A cross-section of the model will be found in Fig. 14 (Plate III). In the diagram, *a* indicates the tank, filled with water up to *b b*; *c* is the model ship, and *d* the plate spring, placed on an intermediate shaft and representing the impulse of the waves.

The intermediate shaft, by means of link *e*, is connected to the crank gear *f*, which can be moved either by an electro-motor or a handbrace, with an intermediate toothed gear. The oscillations of the model are recorded by a pencil fixed at the end of link *k*, turning round the point *i*. The oscillations are marked on drum *l* driven by an electro-motor.

Similarly, the impulses imparted to the model by the plate spring are recorded on the same drum by a pencil fixed at the end of rod *n*, which is placed in a bearing at the pivot of the plate spring lever. The transmission has been calculated, and the pencil placed so that the curves recorded will mark the magnitude of the impulses as well as their phase in relation to the oscillation of the model.

The model floats in the tank, but is kept so that it can only roll and pitch, to the exclusion of all sideways movements. To this end the model has been fitted fore and aft with pivots *o*, which are allowed to slide in a vertical slot. Also, on the model ship the pivots may be shifted up or down, in order to find the so-called axis of oscillation, *i.e.*, the axis about which the ship will be more or less likely to roll when at sea. This axis cannot be ascertained by calculation, as it will always depend on the distribution of the cargo in the ship, and on the underwater shape and the disposition of the appendages.

The true axis, however, may be supposed to be the one about which the ship will roll with the least resistance. In order to ascertain the true position of this axis extinction curves were recorded, with the said pivots on the model at different levels. The model was then heeled to a certain point and left to itself; by means of graphic diagrams it could thus be found how many oscillations of the model took place before

the heeling angle was reduced to a certain point. The axis which during these experiments required the maximum number of oscillations was kept as the standard one for the subsequent experiments. From the experiments it has been found that for most ships the axis of rolling practically coincides with the center of gravity.

In the experimental tank the principal types of vessels (merchant ships as well as men-of-war) have been studied systematically as regards their behavior with the anti-rolling tanks in and out of action. In amplification of the experiments the tanks have been frequently altered in shape and dimensions and also moved about, in order to find out the limits of their efficiency. The data thus obtained were very ample and interesting, but their publication *in extenso* would lead too far. As an example, the particulars of one experiment may, however, be given.

On Plate IV there will be found in Figs. 15 and 16, the results obtained with the model of an ordinary merchant vessel. In the following table the principal data are given for the model and for the corresponding ship, which has very similar proportions to the *Ypiranga*.

	Model. $\frac{1}{48}$ full size.	Ship.
Length between perpendiculars.....L	10.75 feet.	430 feet.
Extreme breadth.....B	1.38 feet.	55 feet.
Draught .....T	.59 foot.	23.6 feet.
Metacentric height.....MG	.91 inch.	36.4 inches.
Displacement.....P	407 pounds.	11,600 tons.
Radius of inertia .....K	.55 foot.	22 feet.
Oscillations per minute.....Z	27	4.27
Weight of water in the anti-rolling tanks....	5.3 pounds.	152 tons.

The anti-rolling tanks used during the experiments hereafter described were placed so as to resemble as closely as possible those constructed afterwards on S.S. *Ypiranga* and her sister ship *Corcovado*. During the experiments, as well as later on board of these vessels, the tanks were put in and out of action by simply opening or blocking the upper air connection of the wing tanks. After the air connection has been closed the free passage of the air enclosed in the tanks is cut off. The water

when oscillating, is, therefore, working against a kind of air buffer, and the oscillations will be reduced to very slight movements. On the other hand, by opening the air connection the water is set free again.

For the experiments the ship was supposed to be struck by waves of different energy and parallel to her longitudinal axis. The angle α (see Fig. A) of the theoretical maximum slope of

FIG. A.

wave was regarded as a measure of the energy. Waves which strike with uniform slope against the whole length of a ship are practically unknown, and are to be considered as merely theoretical, in order to obtain a measure for the impulses imparted to the ship by the waves. Their relation to the real wave slope, which, in some cases, will reach 8 degrees to 9 degrees, will be found in Appendix II.

Model experiments have been made with the following theoretical slopes: $\alpha=1°$, $2°$ and $3°$. The crank radius of the plate spring moving gear was accordingly adjusted by continually varying the radius, until the heeling angle of the model at the farthest range of the spring corresponded to the respective wave slope. Theoretical considerations prove that the impulses to and fro, imparted to the model by the spring, must have the same oscillatory effect as waves of corresponding slope, if only the periods be equal.

First of all, the behavior of the model was noted, when being struck by a series of waves of equal periods; the periods being varied from 32 to 18 per minute for the model (corresponding to 5.06 to 2.85 for the ship). During these experiments the model, first kept in the central position for each series of waves, was set free and the gradually increasing oscillation curves were recorded, beginning from the vertical position. The curves of extinction were also taken down, *i.e.*, the curves of the decreasing oscillations of the model when returning from a certain point of heeling to its normal position.

It is interesting to compare the behavior of the model at the different slopes and periods of the waves. With the anti-rolling tank out of action, for every wave slope, and especially in the proximity of resonance, *i.e.*, co-periodicity between wave and model (27 per minute=4.27 for the ship) from oscillation to oscillation, an increment of heeling is observed, the increase being nearly proportional to the energy of the waves. Above and below this resonance point the increment of rolling was in all cases far less.

From theoretical reasons it appears that the increase of rolling from oscillation to oscillation in case of resonance ought to be nearly proportional to the slope of the wave. This is confirmed by Fig. 16 (Plate IV), showing the curves of increment of heeling for different slopes of waves at resonance. In the same diagram theoretical curves (shown in dots and dashes) have been inserted, showing the line which would be followed by the oscillations at the different wave slopes, if the ship or model could swing in the water without any friction or form resistance.

These lines have been constructed on the theory that the increment of heeling range from double oscillation to double oscillation ought, for each degree of wave slope, to be $=\pi=3.14$ degrees. Thus, for a wave slope of 2 degrees the increment ought to be $3.14\times2=6.28$ degrees, but the real increase, owing to friction and resistance of the ship against rolling motion, is less. In the case of a wave slope of 2 degrees the increment of heeling between the first and second impulse is 4.6 degrees, between the second and third=4.3 degrees, and between the third and fourth=4 degrees.

Through augmentation of the hull's resistance, as by adding bilge keels, the increment of heeling can be further lessened. This point also has found due consideration in the experiments.

There remains the question whether the laws of similarity between model and ship also apply to the resistance of the hull against rolling motion. Opinions may differ on this point, but it can be assumed that the law applies to the ship when at

rest with sufficient accuracy.* For the moving ship, however, this does not seem to be the case with absolute correctness.

It will be possible to make further use of these data by obtaining oscillating curves on board ships when at sea, in order to ascertain the values of real increment. The latter, compared with the corresponding data obtained on models, will enable the necessary deductions to be made as to the energies of the impulses of waves for each case, in degrees of theoretical slope.

Up to now we have only spoken about model experiments with the anti-rolling tanks out of action. The same experiments have also been made with the tanks acting, and the results thus obtained are shown in Fig. 15, immediately below the curves of the former experiments.

Near the dangerous point of resonance the following results have been obtained for the heeling after the fourth impulses of equal period (see Fig. 16):

Slope of waves.	Damping of rolling motion.	With both tanks on.	With only one tank on.
degrees.	*degrees.*	*degrees.*	*degrees.*
1	From 8.4 upwards.	1	2
2	17.8 "	2.7	7.8
3	26.8 "	6.2	17.5

As will be noted, the damping action of both tanks at the same time is almost perfect up to wave slopes of 2 degrees, and decreases slightly only at 3. With only one tank in action the efficiency is far less, especially at a slope of 3 degrees. This proves that with the dimensions assumed for the anti-rolling tanks, one of them by itself would not be sufficient for resisting strong impulses, which, however, happen but rarely.

* This is proved by the experiments carried out some time ago in England on big ships and the corresponding models, in order to ascertain the extinction curves. It was then found that the decrease of heeling from oscillation to oscillation was identical, both for model and ship. This admits of the deduction that the damping action of the hull's resistance (including bilge keels) also follows the laws of analogy.

The curves showing the behavior of the model under impulses above the resonance point, *i.e.,* impulses of shorter periods, as a rule, also show a considerable damping of the rolling for all slopes. This refers to impulses below the said point, viz: the longer periods, but it is to be noted that very slow impulses will show a small increase of heeling.

This may be explained as follows: As already stated, for the model experiments the energy of the impulses was ascertained by varying the crank radius, until the model at the greatest deflection of the plate spring heeled at an angle corresponding to the respective slope of wave. Now these experiments were made with closed air connections of the anti-rolling tanks, which put the latter out of action. This method is only correct near the point of resonance; above it the impulses so obtained prove a trifle too weak, while they are a good deal too strong below that point.

The tank water, when set free, will in the former case lessen the action of the impulses, while it will considerably increase this action below the point of resonance. This is due to the varied phases of oscillation between impetus of wave and ship as well as between ship and tank water. These phases come to 90 degrees at the point of resonance, whereas they will be 180 degrees above and 0 degrees below that point.

With the same model a series of experiments was made in order to ascertain its behavior towards irregular impulses. During these experiments the curves of oscillation were recorded for constant wave slopes but changing periods. The latter varied from 15.8 to 68.5 per minute for the model =2.5 to 11 for the ship.

In Fig 17 (Plate IV) the curves are given for wave slopes of 1 degree, 2 degrees and 3 degrees, both with the anti-rolling tanks in and out of action; the efficiency of the tanks, especially at the high slope of 3 degrees, will be noted. During these experiments the electro-motor was uncoupled and the plate spring was made to move irregularly by a crank worked by hand. The line indicating the curve of irregular impulses was recorded by the writing apparatus, connected with the

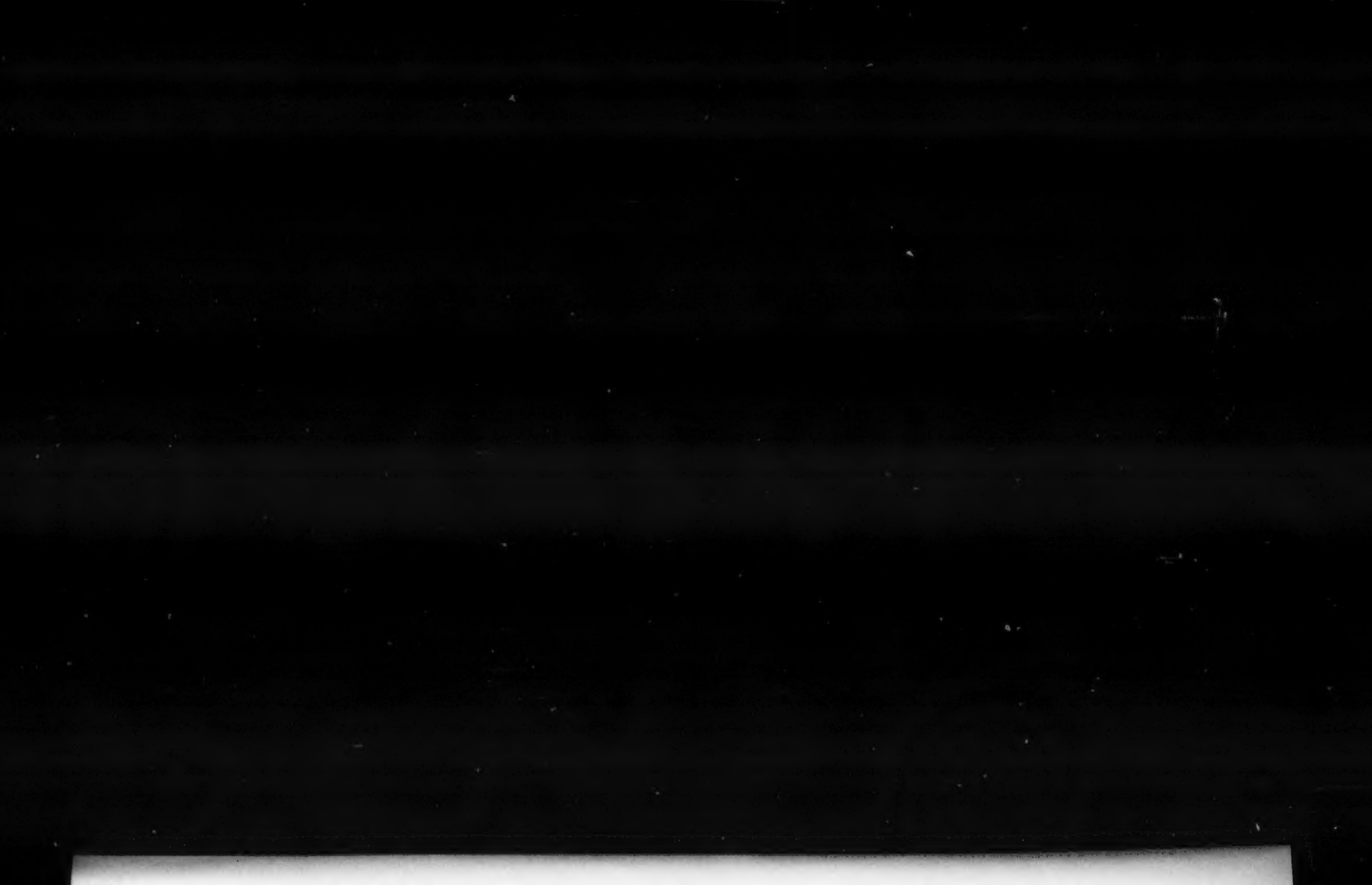


Fig. 1

TWIN-SCREW CARGO AND PASSENGER-STEAMERS "YPIRAN

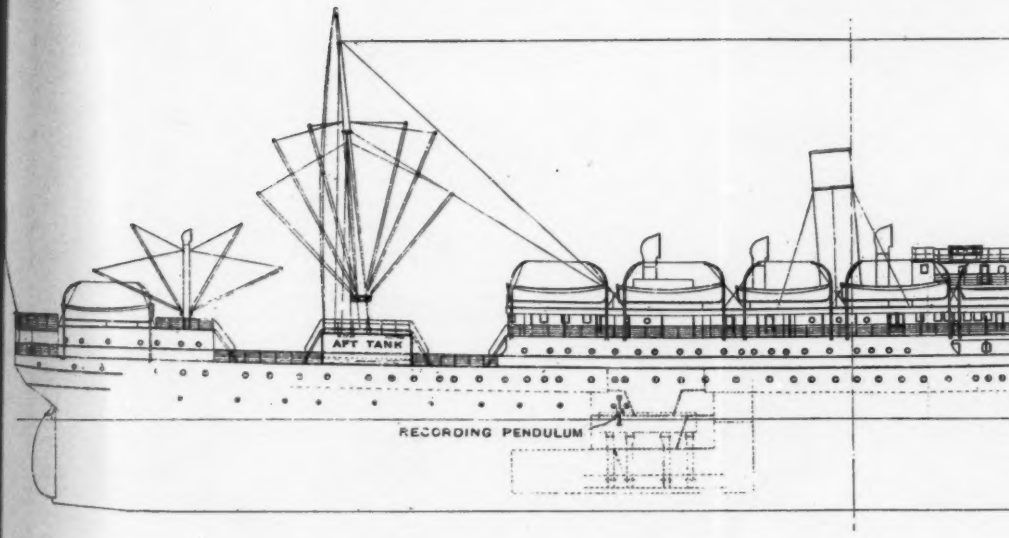
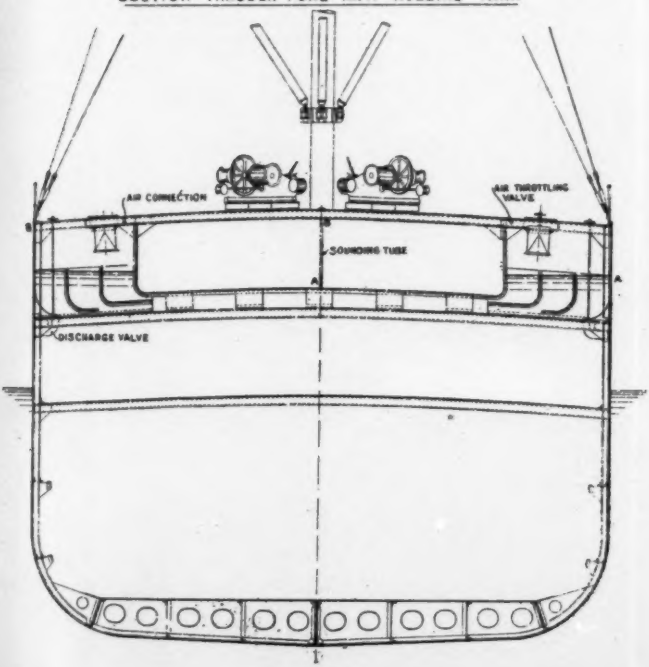


Fig. 2.

SECTION THROUGH FORE ANTI-ROLLING TANK



SECTION

1

AMERS "YPIRANGA" AND "CORCOVADO."

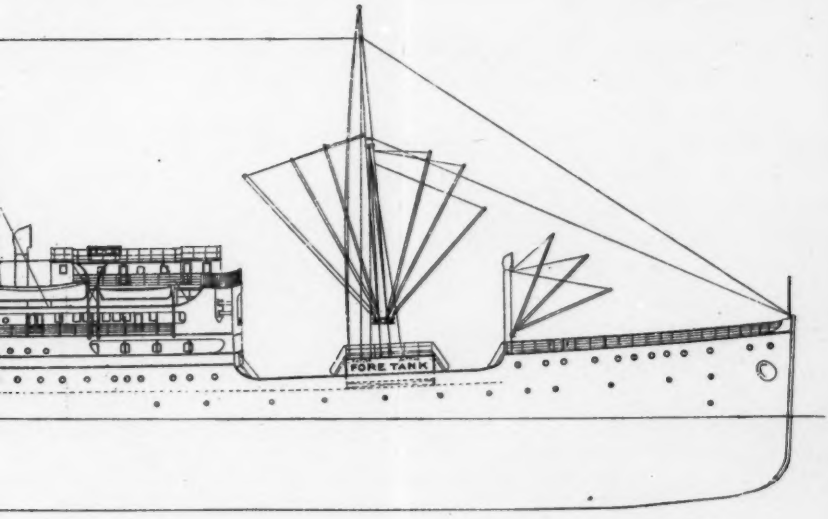
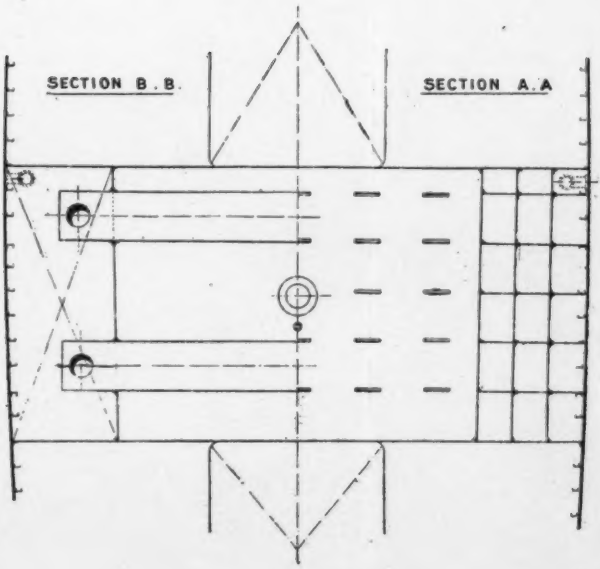
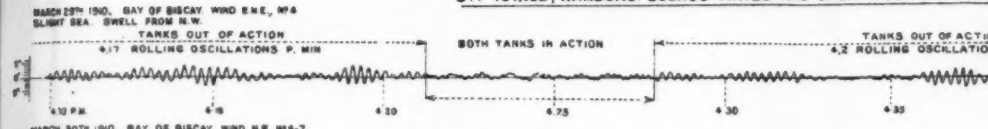


Fig. 3.

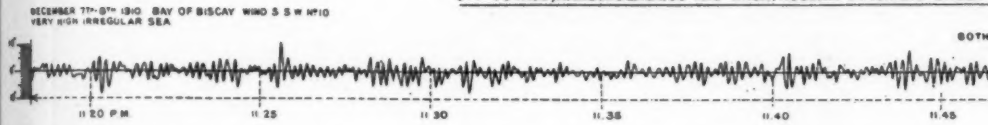


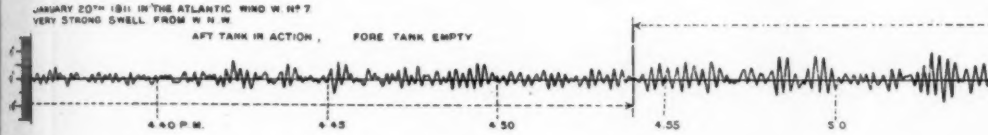
S . S .

5TH VOYAGE, HAMBURG. BUENOS-AIRES AND BACK. ROLLING DIA



9TH VOYAGE, HAMBURG_MEXICO AND BACK. ROLLING DIAGRAMS UNDER





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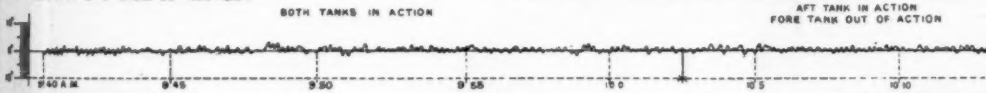
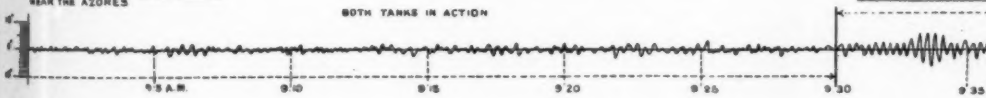




10TH VOYAGE, H

10TH VOYAGE. HAMBURG_MEXICO AN

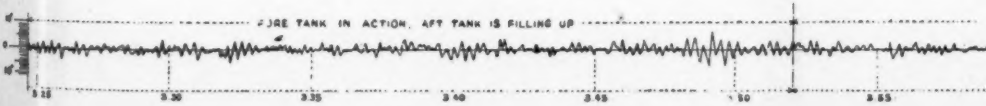


Fig. 4.

."YPIRANGA

DIAGRAMS RECORDED UNDER AVERAGE CONDITIONS OFTEN TO BE MET WITH

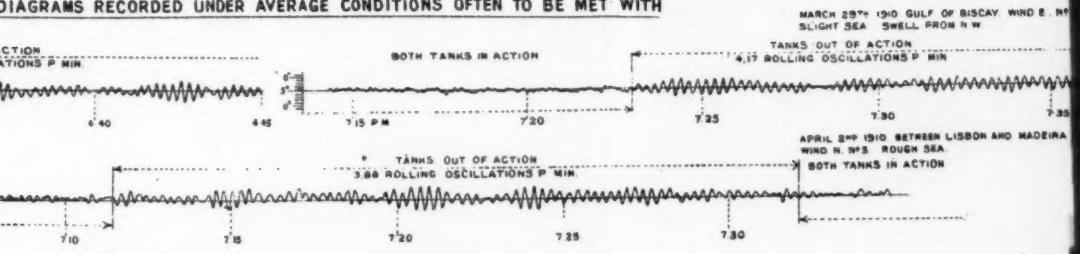


Fig. 6

ER VERY UNFAVOURABLE CONDITIONS (HURRICANES AND EXCEEDINGLY HEAVY SEA)

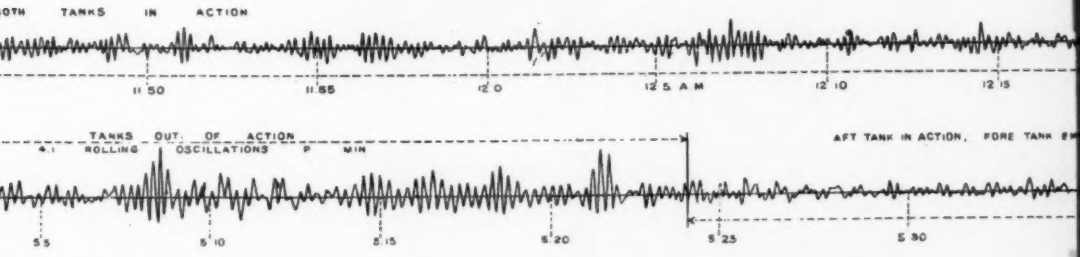
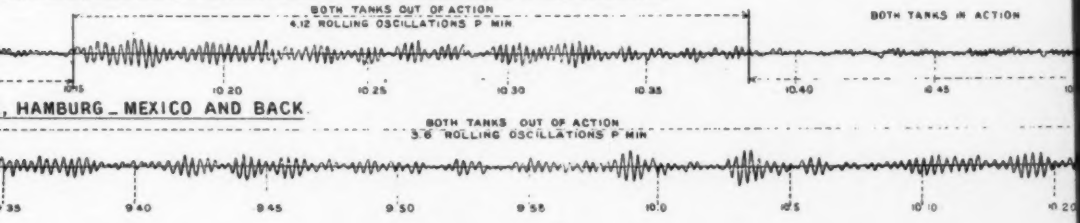


Fig. 5.

S. "CORCOVADO"

NG DIAGRAMS RECORDED UNDER AVERAGE CONDITIONS OFTEN TO BE MET WITH.



, HAMBURG_MEXICO AND BACK.

Fig. 7

AND BACK. ROLLING DIAGRAMS UNDER VERY UNFAVOURABLE CONDITIONS.

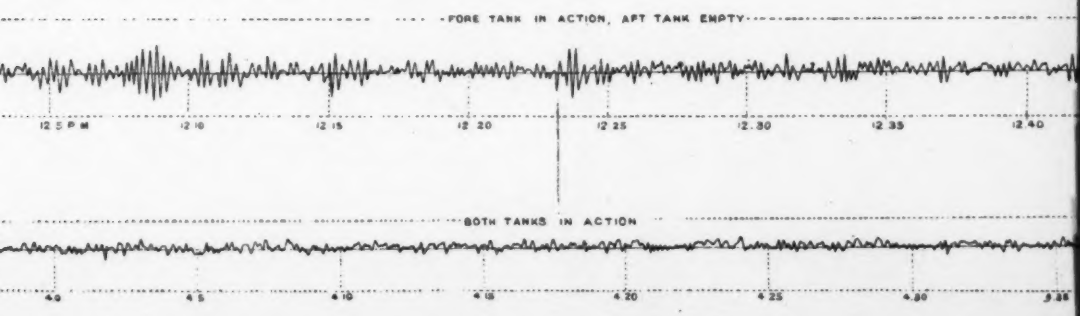


Fig. 4.

S. S. "YPIRANGA

5TH VOYAGE, HAMBURG. BUENOS-AIRES AND BACK. ROLLING DIAGRAMS RECORDED UNDER AVERAGE CONDITIONS O

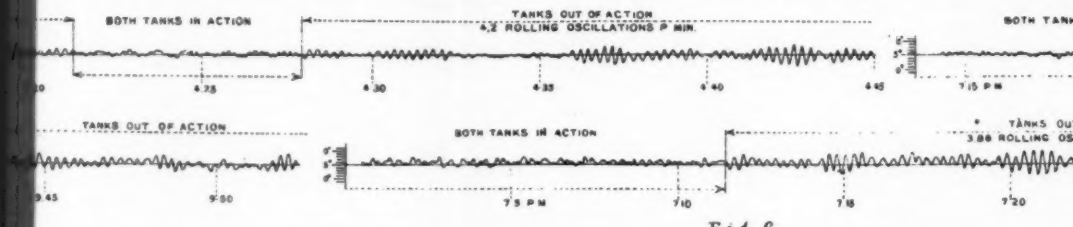


Fig. 6

9TH VOYAGE, HAMBURG_MEXICO AND BACK. ROLLING DIAGRAMS UNDER VERY UNFAVOURABLE CONDITIONS (HURRICANES AND E

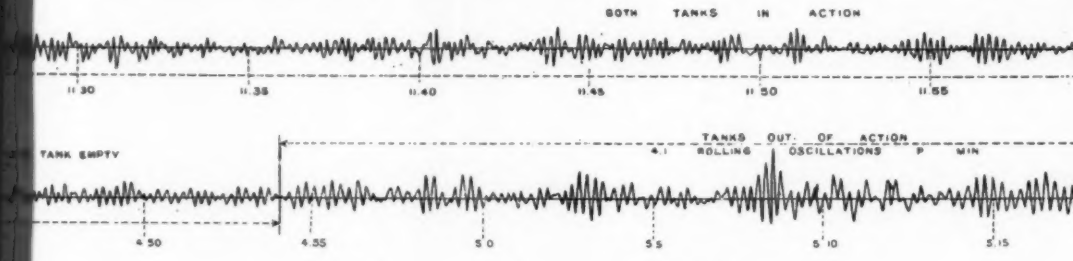


Fig. 5.

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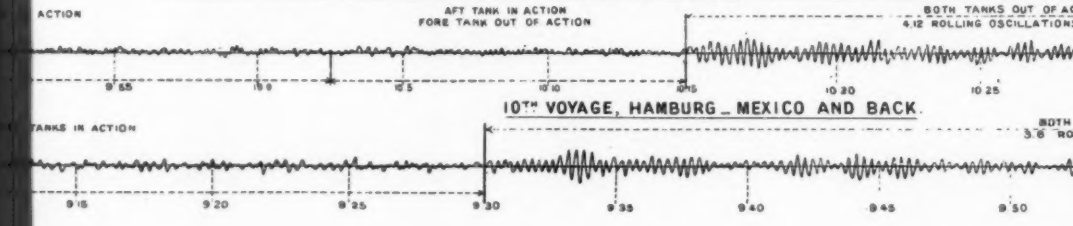
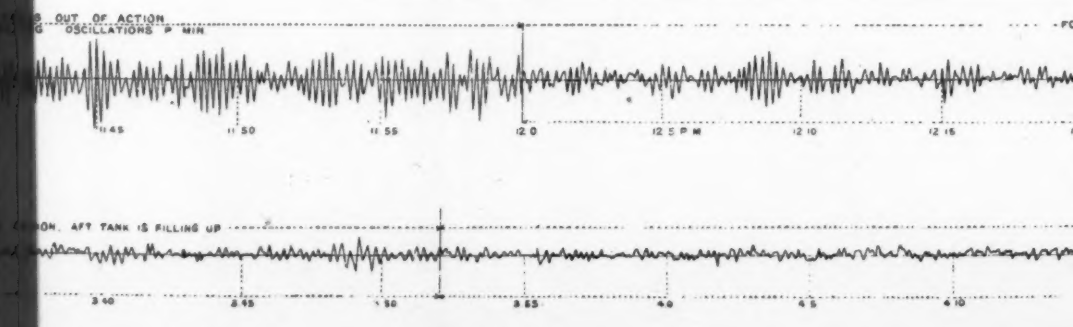
8TH VOYAGE, HAMBURG BUENOS AIRES AND BACK. ROLLING DIAGRAMS RECORDED UNDER AVERAGE CONDITIONS O



10TH VOYAGE, HAMBURG_MEXICO AND BACK.

Fig. 7

10TH VOYAGE. HAMBURG_MEXICO AND BACK. ROLLING DIAGRAMS UNDER VERY UNFAVOURABL



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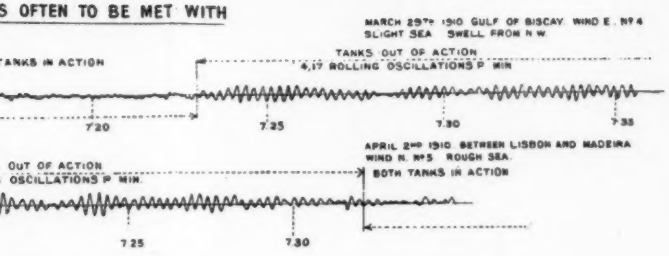

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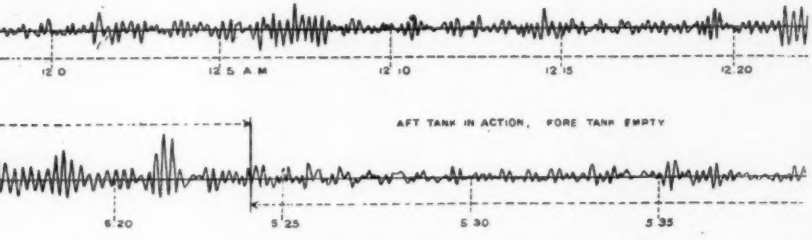

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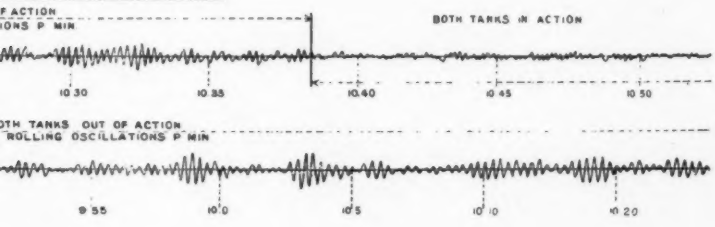

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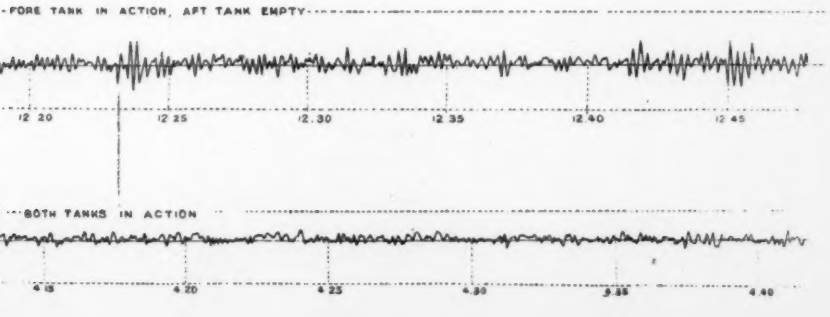


Fig. 8.

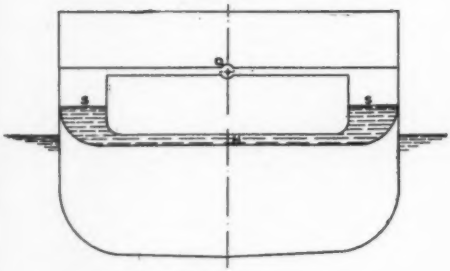
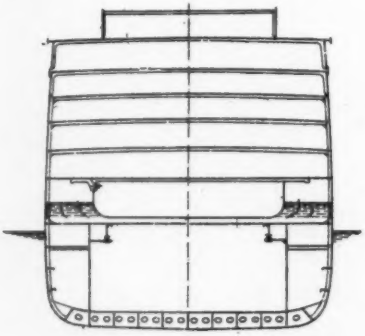


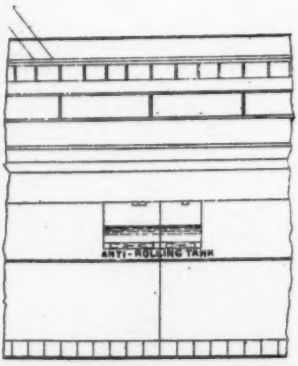
Fig. 9.

LARGE PASSENGER AND CARGO STEAMER.

LENGTH BETWEEN PERPENDICULARS	840'
BREADTH	98'
DRAFT	33'. 6"
DISPLACEMENT	51000 TONS



SECTION THROUGH ANTI-ROLLING TANK.



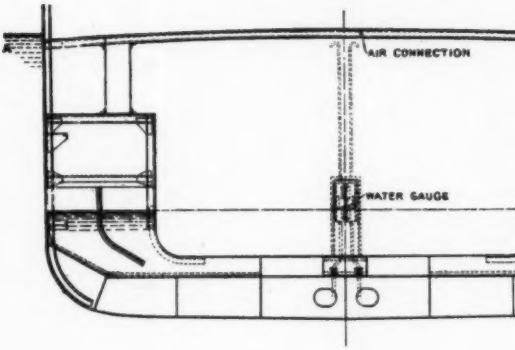
PROFILE.

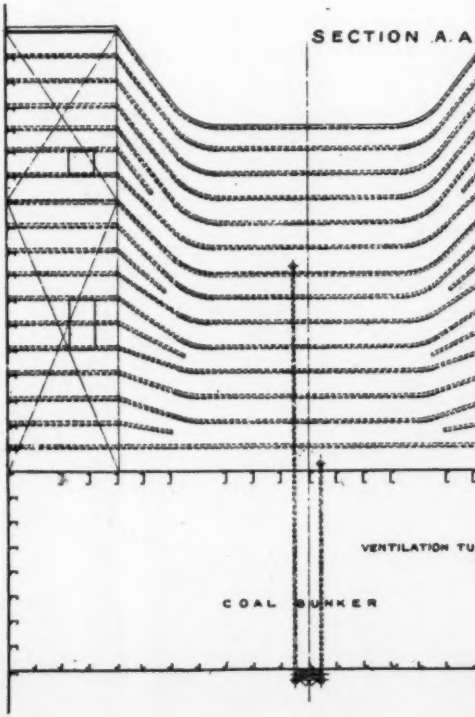
Fig. 10.

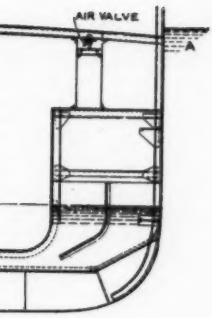
S. S. "GENERAL"

ANTI-ROLLING TANK.

CROSS SECTION.







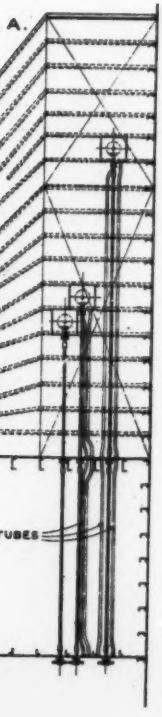
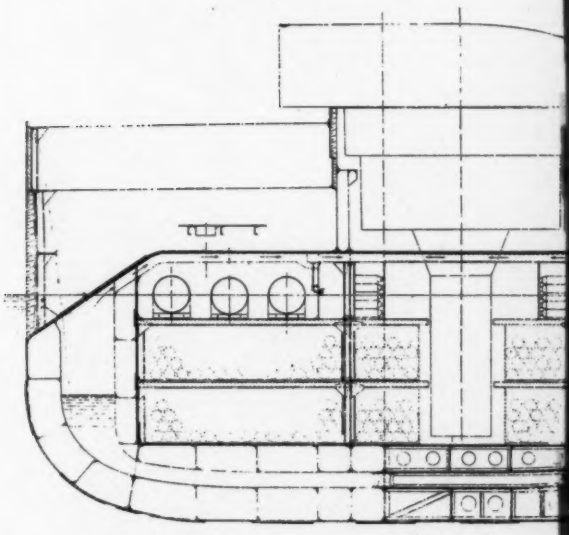


Fig. 11

BATTLESHIP OF 25,600 TONS DISPL



SECTION

Fig. 12.

PROFILE

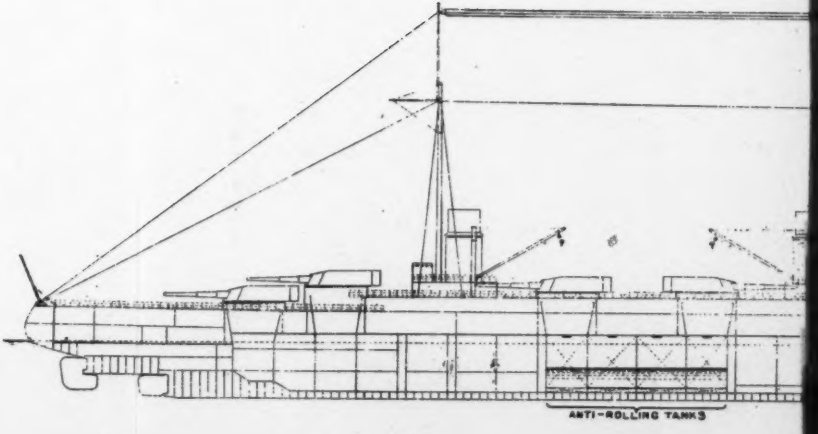
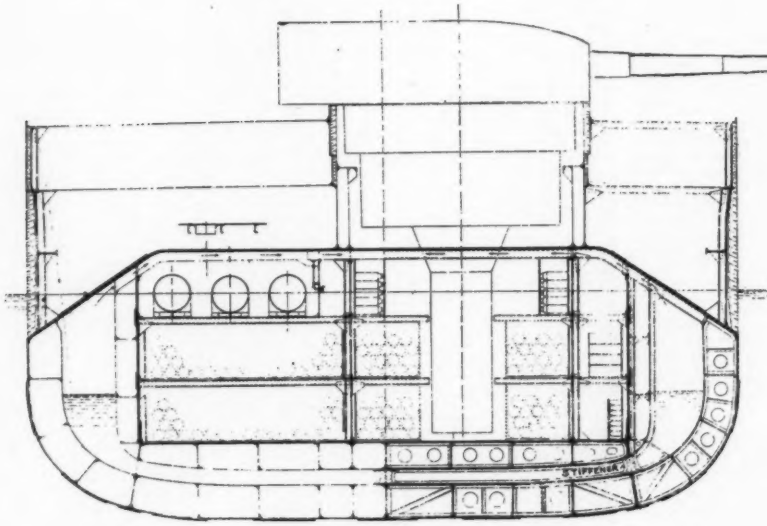


Fig. 11

BATTLESHIP OF 25.600 TONS DISPLACEMENT.



SECTION

Fig. 12.

PROFILE

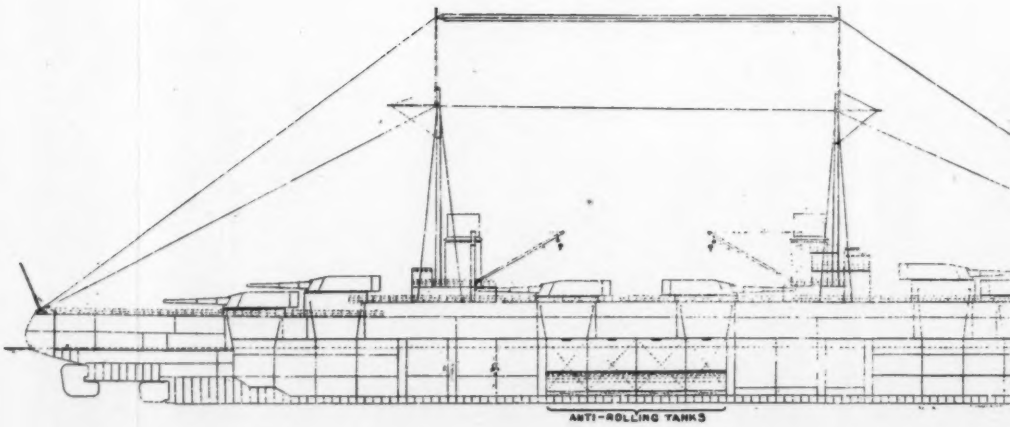


Fig. 13
ROLL PENDULUM

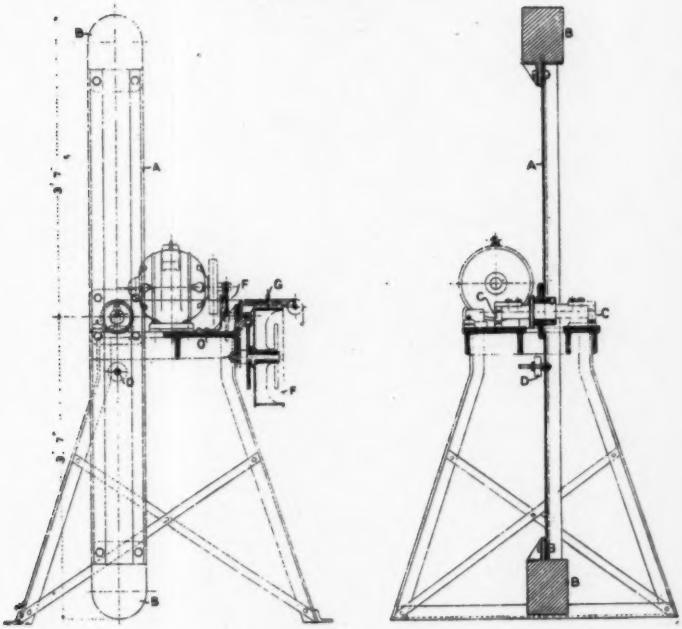
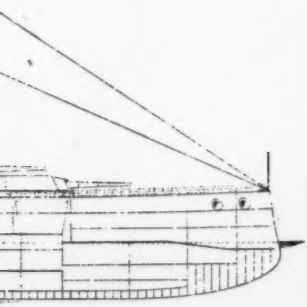


Fig. 14.



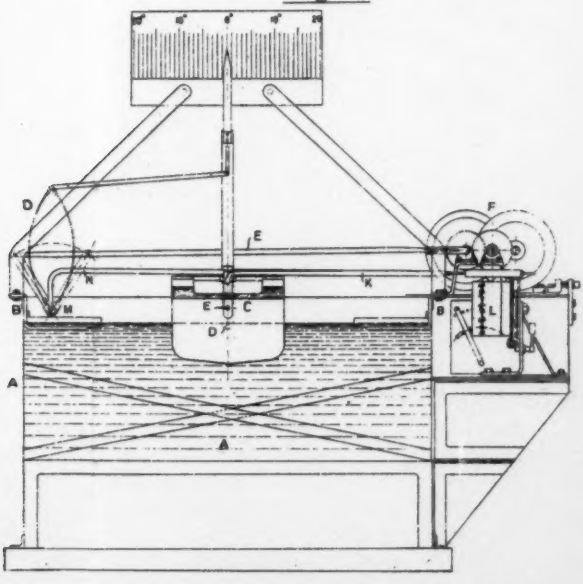


Fig. 15.

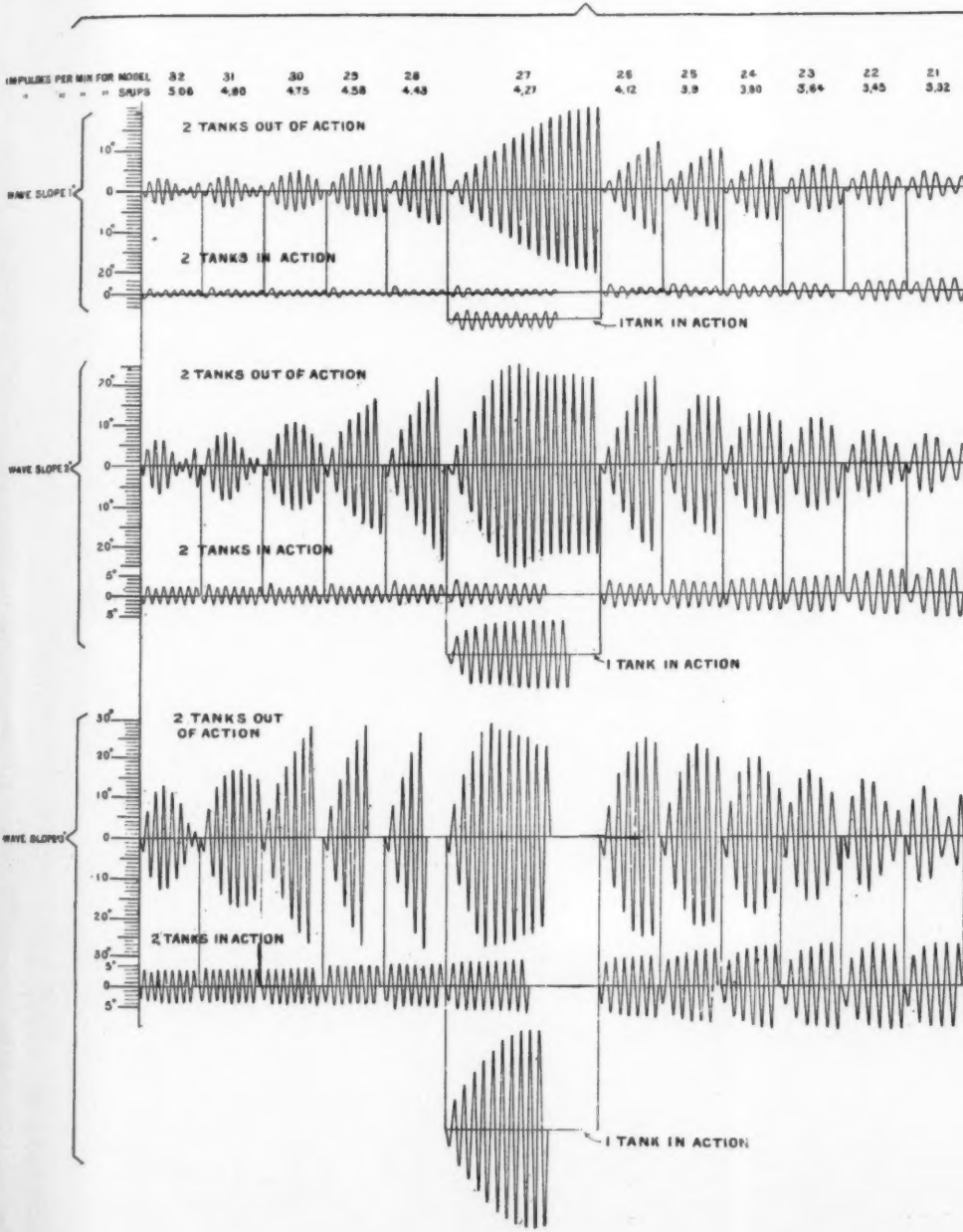
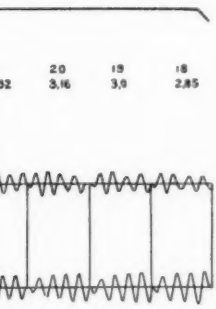
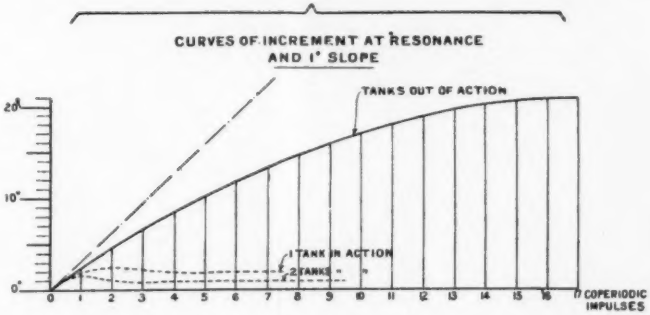
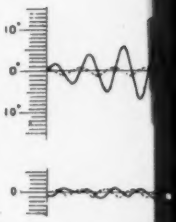
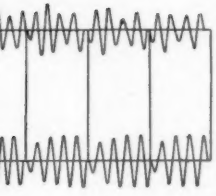

IMPULSES PER MIN FOR MODEL
= " " = " SHIPS
32 5.06
31 4.80
30 4.75
29 4.58
28 4.43
27 4.27
26 4.12
25 3.9
24 3.80
23 3.64
22 3.45
21 3.32
WAVE SLOPE 1°
2 TANKS OUT OF ACTION
2 TANKS IN ACTION
1 TANK IN ACTION
WAVE SLOPE 2°
2 TANKS OUT OF ACTION
2 TANKS IN ACTION
1 TANK IN ACTION
WAVE SLOPE 3°
2 TANKS OUT OF ACTION
2 TANKS IN ACTION
1 TANK IN ACTION

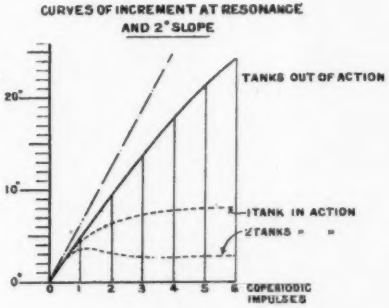
Fig. 16.

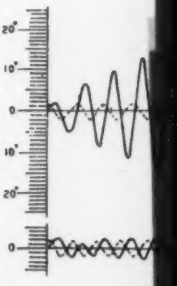


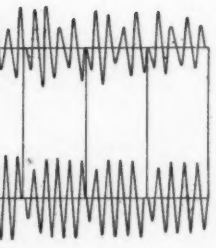

CURVES OF INCREMENT AT RESONANCE
AND 1° SLOPE
TANKS OUT OF ACTION
1 TANK IN ACTION
2 TANKS " "
20°
10°
0°
0 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 COPERIODIC IMPULSES

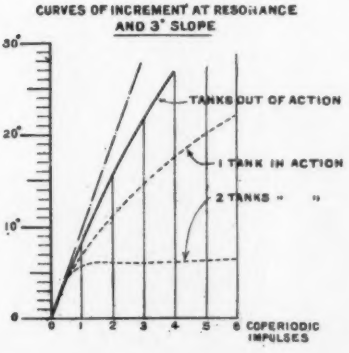


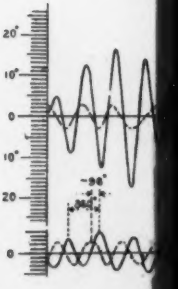






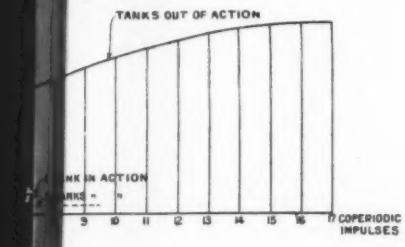






DRAWI
BOTT

MENT AT RESONANCE
° SLOPE



T OF ACTION

ACTION
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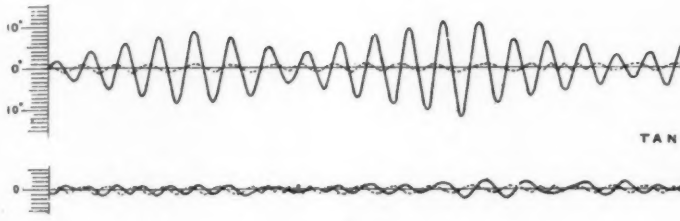
MODE
IMPULSES OF VARIED PERIODS

TANKS



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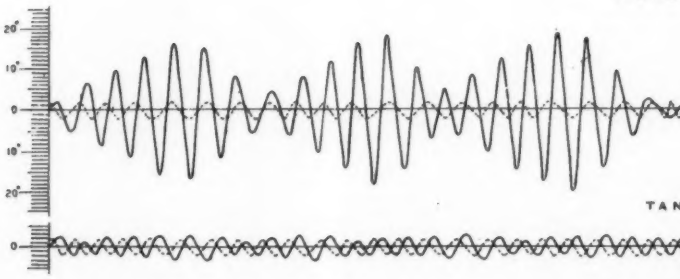
TANKS



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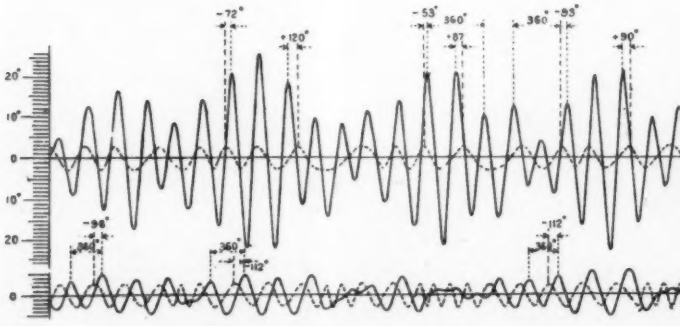
TANKS



TAN

DRAWN LINES · ROLLING MOTIONS OF SHIPS
DOTTED LINES · CURVES OF COURSE OF WAVES

Plate IV.

Fig. 17.

DEL EXPERIMENTS
S AND CONSTANT WAVE SLOPES OF 1°, 2° AND 3°

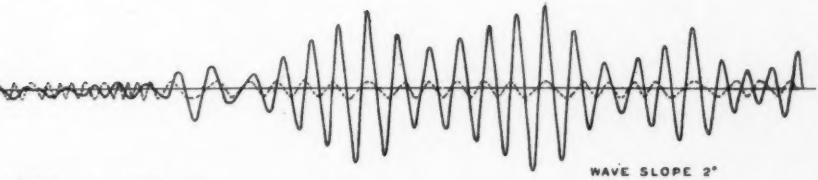
KS OUT OF ACTION

27 ROLLING OSCILLATIONS PER MIN FOR MODEL
4.27 " " " " " SHIPS

WAVE SLOPE 1°

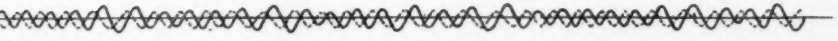
ANKS IN ACTION

KS OUT OF ACTION

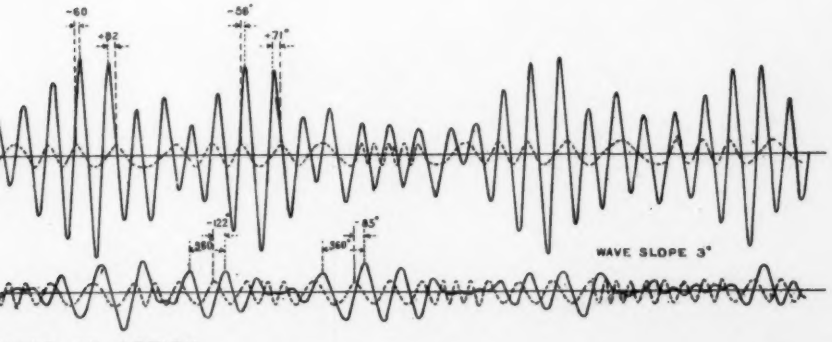


WAVE SLOPE 2°

ANKS IN ACTION



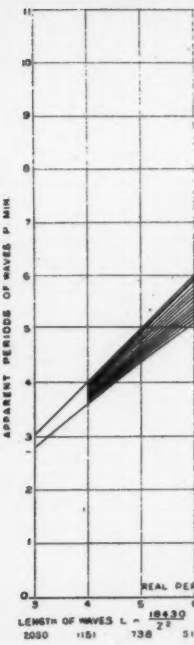
KS OUT OF ACTION

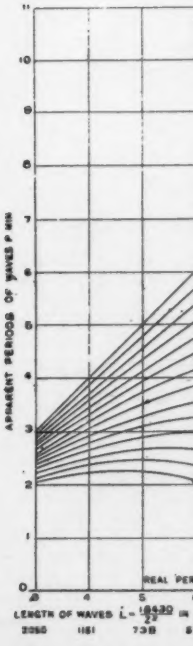


WAVE SLOPE 3°

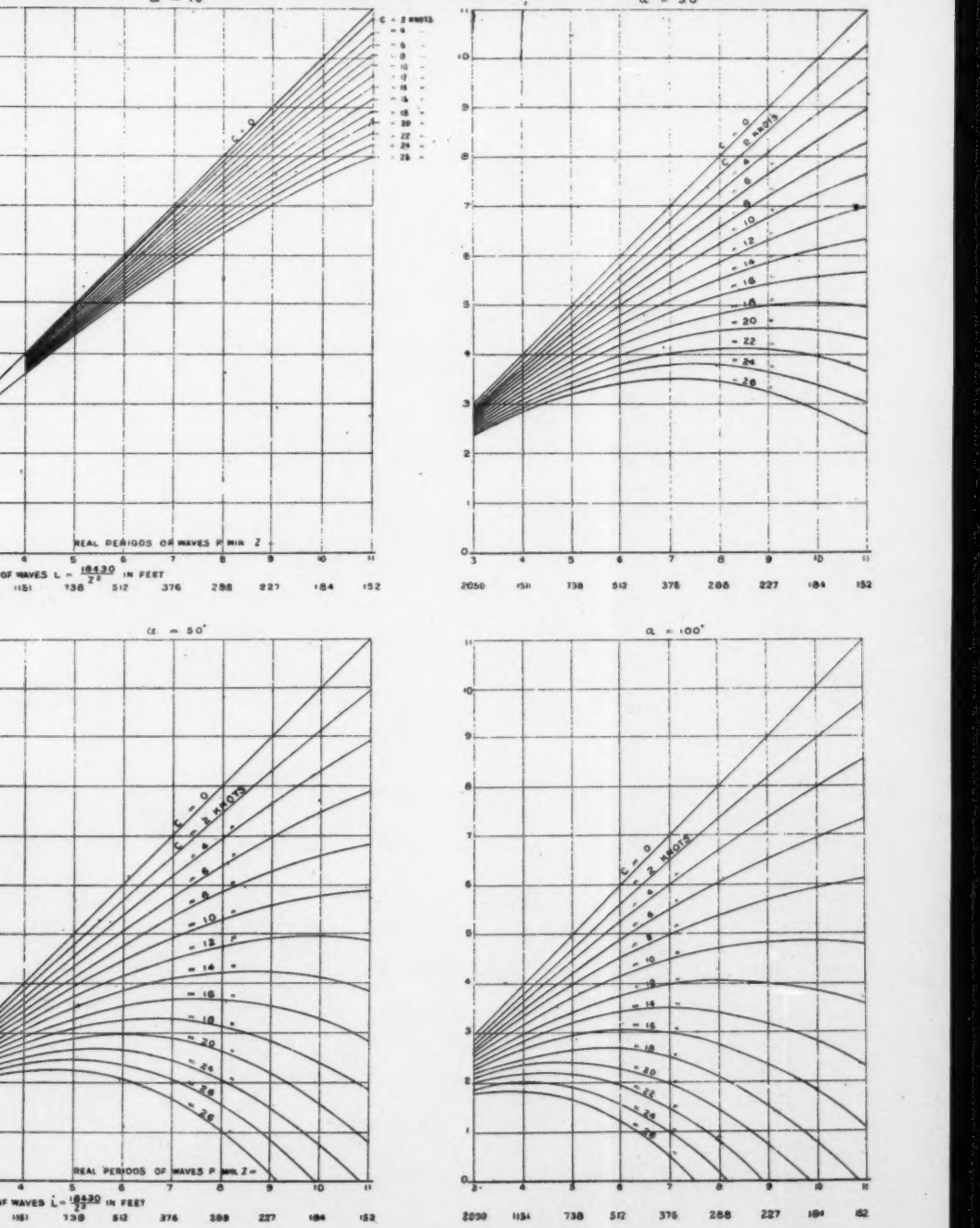
ANKS IN ACTION

TO DEMONSTR


APPARENT PERIODS OF WAVES P MIN
REAL PE
LENGTH OF WAVES L = 18430/P²
2050 1151 738


APPARENT PERIODS OF WAVES P MIN
REAL PE
LENGTH OF WAVES L = 18430/P²
2050 1151 738

TO DEMONSTRATE THE DEPENDANCE OF THE APPARENT PERIODS OF WAVES ON SHIP'S SPEED AND OBLIQUITY OF WAVES.



crank gear, as a separate line and parallel to the oscillation curves. In Fig. 17, however, the curves are illustrated together, in order that the lag of the phases between impetus of waves and ship may be more easily perceived.

It will also be seen that, where the heeling increases, the ship as a rule is 90 degrees ($=\frac{1}{4}$ period) late, while on the other hand she will advance by nearly 90 degrees as soon as the heeling decreases. In the former case the impulse of the waves will augment the oscillations, while it will counteract the ship's movement in the latter case, with a consequent diminution of heeling.

With the anti-rolling tanks in action, on the contrary, the lag of phases remains almost constant, and nearly in all instances the oscillations of the ship will be found about 90 degrees late. As noted in Fig. 17 (3 degrees wave slope) the lag of phase will not always be exactly 90 degrees, but the character of the phase-delay remains.

From the curves obtained it may be deduced that a ship—as has already been mentioned—will roll only in her proper period, even in a very irregular seaway.

The general theory and the method of calculation for particular cases have been published in the Journal of the *Schiffbautechnische-Gesellschaft* for 1911.

APPENDIX II.

GENERAL NOTES ON WAVES AND THEIR EFFECT UPON MOVING SHIPS.

From a large number of observations* it has been shown that the average periods of waves vary from 5 to 11 per minute. The higher periods (8 to 11) are more frequent in shallow waters (North Sea, Baltic, &c.), while the lower are found in deep seas (Atlantic, &c.).

Besides the depth of the water, the period depends upon the

*See A. White, Johow-Krieger, Guyon, etc.

intensity of the wind, its duration, and the extent of water in the direction of the wind. It seems to be certain, however, that at any one time only waves of a certain period are found, *i.e.*, if the general conditions are in favor of waves of period 6, none of 9 or 11 will be noted.

Of course, certain fluctuations are always to be expected, as well as sudden variations of phase, *e.g.*, the waves tilting over. Waves with less than 5 periods per minute are, according to the reports, very rare.

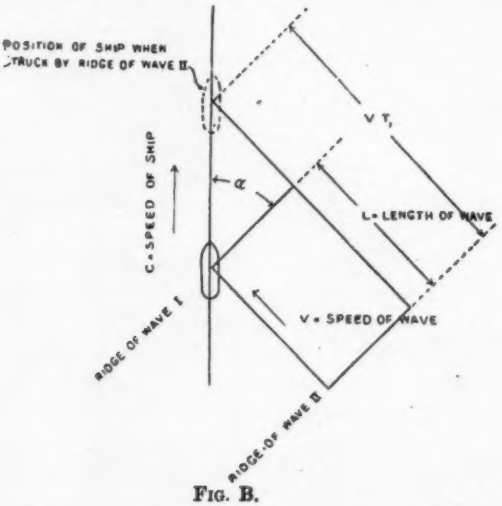

POSITION OF SHIP WHEN STRUCK BY RIDGE OF WAVE II
V T₁
C = SPEED OF SHIP
α
L = LENGTH OF WAVE
V = SPEED OF WAVE
RIDGE OF WAVE I
RIDGE OF WAVE II

FIG. B.

As has repeatedly been stated, real rolling is only produced if the periods of ship and wave synchronize. It must, however, be pointed out that by "wave-period," not its *real period*, but its *apparent period* relative to the moving ship is to be understood. The real period only refers to a stationary vessel, or a ship that moves in a direction exactly parallel to the waves. The apparent period is produced in the following way:

If the vessel proceeds at an angle towards the generating line of the wave, the intervals between the single ridges striking

the ship will be shortened or deferred. They will be shortened if the waves come obliquely from ahead, and lengthened if they come up from behind (Fig. B).

The apparent period is given by the formula:

$$T_1 = \frac{L}{v - c \sin a}.$$

In this formula:

T_1 =apparent period of wave in seconds.

L =real length of wave in feet.

v =real velocity of wave in feet per second.

c =speed of ship in feet per second.

a =angle between the ship's longitudinal axis and the generating line of wave. This is positive if the waves come up from behind, and negative if from ahead.

Besides, there exists the relation: $L = v \cdot T_1$, where T signifies the real wave-period. Further, according to the general wave theory:

$$L = 5.12 \cdot t^2 = 5.12 \left(\frac{60}{z}\right)^2 = \frac{18{,}430}{z^2},$$

z signifying the number of real period of the wave per minute.

In Plate V, for periods of 3 to 11 per minute the corresponding apparent periods for different speeds up to 26 knots at angles of obliquity of 10 degrees, 30 degrees, 50 degrees and 70 degrees, are shown, in order to demonstrate how these figures depend on the ship's speed and the obliquity of the waves. The waves for these diagrams have been supposed to come up obliquely from behind.

Waves coming in at an angle from ahead have not been especially treated, as these waves increase the apparent number of period, the latter thus getting quite out of the range of the individual phases of big ships, which interest us more especially.

The curves thus obtained are interesting from different

points of view. We find, for instance, that with fast vessels and very oblique waves coming up from behind, the number of apparent periods of the waves will suffer such a marked decrease that this number may be less than half of that of the real periods.

If, for instance, the waves have 8 periods per minute, the apparent period at a speed of the ship of 26 knots will be at an—

Angle of 20 degrees, about 4.8 per minute.
Angle of 30 degrees, about 2.4 per minute.
Angle of 40 degrees, about 2.1 per minute.

Now, if the individual periods of oscillation of the vessel coincide with one of these values considerable rolling would be likely to ensue on account of resonance.

This will also explain the well-known fact that the big fast liners (including the Cunarders), though their individual period of oscillation is very low, and sometimes even below 3 per minute, will roll severely in comparatively short waves, coming in at an angle from behind.

In general it seems to appear from the curves that nearly every ship, in consequence of her speed, and, notwithstanding her period, may, under certain circumstances, have to deal with conditions where the impulses of waves and her individual oscillations synchronize, and thus cause her to roll. The requirements for this to happen are naturally very different.

In order to obtain a better idea of the heeling action of oblique waves upon a ship, graphic researches assuming actual wave slopes of 8 degrees to 9 degrees have been made, with the result that for big ships it seems hardly necessary for calculation purposes to go beyond a maximum theoretical slope of $\alpha = 2$ degrees to 3 degrees. This is confirmed by a number of practical observations (see rolling diagrams of S.S. *Ypiranga* and *Corcovado*), from which it results that an increment of rolling of 5 degrees to 6 degrees for one double

oscillation, corresponding to a theoretical slope of 2 degrees to 2.5 degrees, is to be met with but rarely.

For the relation between the increments of rolling and the wave slope, see Appendix I.

This small theoretical slope is to be explained by the fact that, in consequence of their irregular shape, the waves, even if coming in athwartship, will strike big ships only partially. For waves coming up obliquely from behind the theoretical slope will be still less.

THE *MINNESOTA'S* TIME-FIRING DEVICE MADE ON BOARD SHIP.

By Lieutenant John J. Hyland, U. S. N., Member.

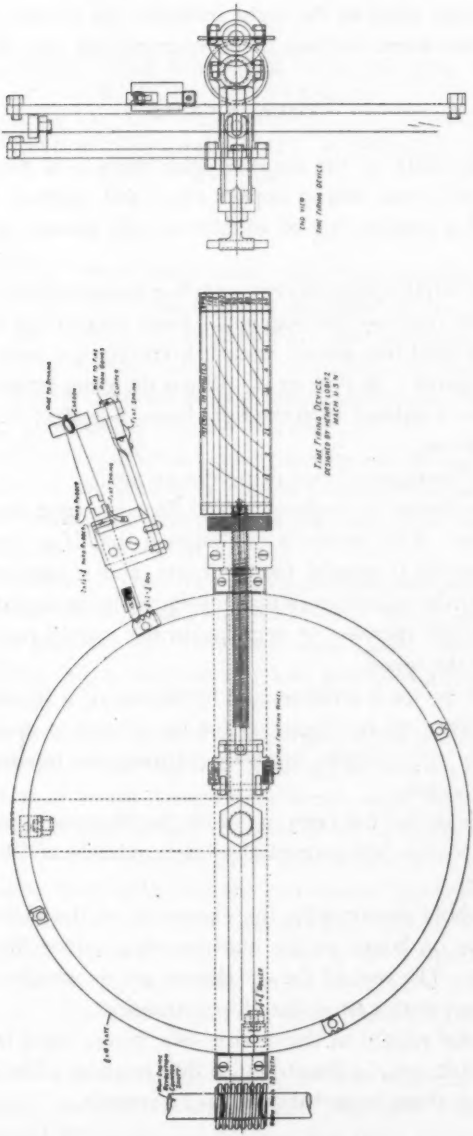
The following description applies to the Lobitz time-firing device, as designed by Machinist Henry Lobitz, U. S. Navy. The device was made and installed by the ship's force on board the *Minnesota*.

Referring to the accompanying figure, there is a small leather-bound friction wheel, 2½ inches diameter, placed on a shaft driven from the main-engine revolution-indicator gear. The wheel is driven through worm gearing proportioned to give one revolution of the wheel for two hundred revolutions of the main engine.

The friction wheel makes contact with the face of and drives an 18-inch disc as shown. The friction wheel is loosely mounted on its shaft, with a sliding feather for driving it and with an adjusting screw for varying its distance from the center of the disc. The pressure between the face of disc and circumference of friction wheel can be varied by an adjusting nut at the center of the disc. This nut has a ball-bearing under it to lessen friction.

GONG SIGNALS.

The circumference of the disc carries four lugs spaced 90 degrees apart. These lugs carry one, two, three and four teeth, respectively. As the disc revolves each tooth in turn engages the contact maker, shown at the right in the figure, and closes a circuit leading to an electric gong in each fireroom. The gongs are thus made to strike one, two, three and four times at regular intervals, the length of the intervals depending upon, first, the speed of the main engines, and, second, on the distance of the friction wheel from the center of the disc.



The index plate at the right indicates the proper setting of the friction wheel for any firing interval that may be desired.

VISUAL SIGNALS.

On the shaft of the rotating disc there is a 6-inch hard-rubber disc with, first, a copper ring, and, second, four segments of a copper ring at equal intervals around its circumference.

On the friction disc there is a carbon holder which rides over the copper ring and the segment. Each segment is wired to a lamp in a light box placed in each fireroom, the light box having the figures 1, 2, 3, 4 on it. When the gong strikes one, the figure one is lighted up in the light box—2 gongs; No. 2, lights up, and so on.

The advantages of this firing device are:

1. The timing is mechanical and depends upon the speed of the engine. The device is so designed that if a three-minute firing interval is needed for 10 knots, and if the speed is increased or decreased, as in maneuvering, the firing interval will automatically increase or decrease in the correct proportion to maintain the speed.

2. The device is positive, and by the use of a small constant-speed motor, can be rigged up for use in port if desired.

3. It is very simple, is easily adjusted or repaired by the force on watch.

4. This device has been in use on the *Minnesota* for the past sixteen months, has proved absolutely reliable and has needed no repairs.

The whole device with the exception of the electric gongs was made on board in the machine shop under Mr. Lobitz's direction. The cost of the six electric gongs was $45.00. The light boxes were cast in the ship's foundry.

The total weight of the device—six gongs, light boxes, wiring, conduit, etc., is about 200 to 300 pounds. The weight of the device alone is probably 50 to 75 pounds.

SOME REMARKS ON THE *DELAWARE* AS A CRUISER BATTLESHIP.

By Lieutenant Commander H. B. Price, U. S. N., Member.

Note.—The recent remarkable performance of the *Delaware* makes the following account, prepared by her chief engineer, at the request of the editor of the Journal, of very special interest to our readers. It is needless to remark that no small share of the credit for this performance belongs to Lieutenant Commander Price.—Editor.

There having been some press comment on the recent full-power trials of the *Delaware*, which immediately followed a run to Valparaiso and return, brief mention of a few points in this connection may be of interest.

In the first place, evidently the ship and her machinery were, on the whole, well designed and well built. At all speeds, including the highest, the *Delaware* slips easily through the water, making only a very small wave, and with almost entire absence of vibration.

The hull structure gives a good, rigid support for the engines, evidenced by the fact that, although when coming up the Eastern coast of South America she struck some heavy storms with very rough seas, giving the ship about as much rack and strain as she is ever likely to experience, there were no warm bearings either then or during the subsequent full-power runs.

Having been ordered from Guantanamo Bay to Hampton Roads entirely unexpectedly, to convey to Valparaiso the remains of the late Chilean Minister to the United States, the *Delaware* left Hampton Roads on this mission January 31, 1911.

The run of 4,918 miles to Rio de Janeiro was made at an average speed of 12.92 knots, using 1,922 tons Pocahontas coal. Reached Rio February 16, and left there February 22,

1911. Made the run of 3,832 miles from Rio to Valparaiso at an average speed of 11.18 knots, using 1,293 tons of Cardiff coal. This coal, obtained at Rio, was bad, having much refractory clinker that stuck to the grate bars, often causing them to be dislodged in the effort to loosen it, and often forming solid pieces so large that they could not be gotten out of the furnace doors until broken up within. This was very hard on the fireroom force; but was good drill for them, making ordinary firing with standard Navy coal seem easy. On this run a stop of two and one-half days was made at Sandy Point, to exchange visits of courtesy with the Chilean civil and naval authorities.

Ten days were spent in Valparaiso harbor, and the ship left for home the evening of March 21, 1911. A stop of fifteen hours was made at Sandy Point to obtain fresh provisions. Made the run of 4,012 miles from Valparaiso to Rio at an average speed of 12.82 knots, using 1,493 tons of very good quality Cardiff coal obtained in Valparaiso.

Stopped at Rio April 4 to 10. Made the run of 4,793 miles to Boston at an average speed of 12.5 knots, using 1,956 tons of good quality Cardiff coal obtained at Rio.

As the ship was to have several weeks of quiet after reaching New York, it was proposed to run full power for a few hours on April 24, two days before arrival, to see by trial what bearings had best be overhauled. But on the morning of April 24 there was a storm with a very heavy head wind and sea, which it would have been unwise to run into at high speed. So the proposed trial was not run. The next day, the weather having improved, two extra boilers were lighted, making ten out of fourteen in use, and the blowers were put on to hold an air pressure of 1.9 inches of water in the three firerooms in use. The ship ran thus for two and a quarter hours, at a speed of 20.07 knots by revolutions, the engines developing 19,000 I.H.P. This run was made with throttles and stops wide open, the steam consumption being adjusted to steam production by linking up.

The bearings all ran nicely. The distances above given are

calculated from revolutions of main engines, so that they are the actual distances and speeds through the water, thus eliminating considerations of effects of currents, etc. Off the coast of Uruguay a severe storm was encountered necessitating deviation from the direct course for about half a day, to avoid danger to the steam launches from seas breaking aboard. Even that weather, however, would not have seriously interfered with the working of turret guns; and there was practically no racing or racking of the engines.

The *Delaware* had orders to dock at Boston immediately upon arrival and then go to the New York yard for a month, to have some minor changes in quarters, etc., made in preparation for attending the coronation ceremonies. No one on board had any intimation of a change in these plans until about 5:00 A. M., April 26, when a wireless message was received, directing that, if feasible, the *Delaware* coal at once and immediately proceed on a four-hour full-power trial, followed continuously by a twenty-hour full-speed endurance run. The ship entered Boston harbor at high tide that forenoon, anchoring about 10:30. Four coal lighters were immediately put alongside, coaling gear was rigged, and during the afternoon the ship took 998 tons of ordinary run-of-the-mine coal. Anticipating docking, the reserve feed tanks were nearly empty, and the evaporator plant was secured for scaling and overhauling. Fresh water was requested of the Navy Yard; but about the middle of the afternoon it was ascertained that the Boston Yard had no water barge and could give the ship no water. The four evaporators were started, and made about 90 tons fresh water, which was enough for the trials and subsequent high-speed standardization runs.

The ship was in Boston harbor 22½ hours, leaving next morning, April 27, at 9:00 A. M.—as early as the tide would serve. During this time, no examination or adjustment whatever was made about the main engines or auxiliaries, except the ash expellers. The men not coaling ship concentrated their efforts on cleaning fire sides of boilers and getting ash expellers in working order. It was necessary to keep

fires in two boilers for auxiliaries and evaporators, so they could not be well cleaned. The men, except those on auxiliary watch, were given a chance for an uninterrupted night's sleep.

The ship left Boston harbor under six boilers, the cleaning of the other eight being still in progress. These were lighted as they were ready, and by 2:42 P. M. all fires were built up heavy and the full-power trial was begun, with the engines running 128 revolutions. This was soon improved, as fires were green at the start, and the average for the four hours was 131.02 R.p.m. There is no definitely established point that high on the ship's official speed-revolution curve; but it gives a speed of about 21:85 knots average for the four hours. The run was continued, with 2 inches air pressure in the firerooms, for twenty hours more.

The average speed for the twenty-four hours was 21.32 knots by the last official curve.

The contract speed of the *Delaware* was 21 knots. This performance seems satisfactory, as it represented strictly service conditions, the orders for a full-speed run having come as an entire surprise to everybody, when it was expected that the ship would dock in accordance with orders already on board.

There had been no opportunity to clean the water sides of boilers since leaving Norfolk in January; yet the boilers steamed freely, and for much of the time the forced-draft blowers were run below full speed to avoid having more than the allowed air pressure in the firerooms.

No special preparation has at any time been used in the boilers. Water from each steaming boiler is tested every morning for salt and acid, and lime water is used to preserve a slight alkalinity.

There was not a single hot bearing or any derangement of main engines or auxiliaries during these full-speed trials and the standardization runs, which followed.

On the conclusion of the twenty-four-hour endurance run the ship slowed down to clean fires well while entering Penobscot Bay. This done, she at once proceeded with the high-

speed standardization runs over the Rockland course. Though the depth there is from 45 to 80 fathoms, the engines would make only 130 revolutions with everything wide open and safety valves lifting, while at sea during the full-power run, they made 132 revolutions under the same conditions.

The men stood a watch in three, and there was one fireman and one coal passer for each pair of furnaces, as in ordinary cruising—half the number employed by the contractors on their full-speed trial of the ship. The firing timing device—which is habitually used on the *Delaware*—was set at one minute and thirty seconds; and this rate required 7 to 8 shovels of coal per charge. Furnaces were fired and raked alternately, as is customary.

After twelve hours' steaming the customary routine cleaning of fires was begun; that is, cleaning one fire in each boiler per watch, or a total of fourteen per watch. Considerable speed was lost by failure to begin cleaning fires at least four hours earlier. They were very dirty, and it was necessary to haul everything out of each furnace cleaned.

No fuel oil was burned during these full-speed runs, as it did not seem needful, and it was thought better to run the trials burning coal only as a comparative demonstration of the ship's steaming capability.

The *Delaware* has burned oil under varying conditions, including a full-speed run of four hours, two days' steaming in fleet during maneuvers, burning oil only; has used it during the target-practice period to effect the quick changes of steam generation then wanted, without loss; has burned oil in conjunction with coal; has burned oil only, in port, for several months. Full reports have been made to the Navy Department relative to oil burning under various conditions.

There were several things contributory to this comparatively successful steaming of the *Delaware*. In the first place, as stated above, machinery and ship are well designed and well built. The forced-lubrication system on the main engines saves the bearings from wear if they are kept reasonably well adjusted. This is the great military advantage of forced

lubrication. Saving of oil is a secondary matter, as any ship can easily carry comparatively large amounts of lubricating oil.

On the *Delaware,* the lubricating-oil economy is not marked, and at full speeds there is much waste of oil, which will be improved upon. But the economy in the wear of bearings, and the consequent general readiness and reliability of the ship for service, is very marked.

In the harbors of Rio and Valparaiso, such needed adjustments and overhaulings were done as was compatible with the crew's liberty ashore. It may be stated, both as to machinery and to personnel, that the *Delaware* has been very free from any slackness and from any friction. The men all have a pride in their ship and an interest in their work, and in that lies a big element of successful accomplishment. A co-ordination of effort, a spirit of mutual helpfulness and consideration of one another's problems and difficulties, a careful adaptation of routine, among officers and men of all departments, is a very real element of military efficiency.

During the three months of this South American voyage, the only thing bought in the way of machinery supplies was a few ½-inch rivets, which were found to be unfit to use. Some fresh water was bought in Rio, because it was convenient and it was cheaper to buy it than to use 50-shilling coal to distill it. The only other items of purchase were coal and provisions.

The *Delaware* is fitted to carry an ample supply of lubricating oil for a cruise around the world without renewal.

Her foundry and machine shop, copper shop and blacksmith shop, have been able to do all the ordinary work requisite to make the ship self-supporting in the matter of repairs. A fuel-oil forge is used for the heavier work in copper and blacksmith shops. The foundry uses fuel oil, of course. It has made castings of iron, brass, composition, aluminum, Babbit metal, and solder.

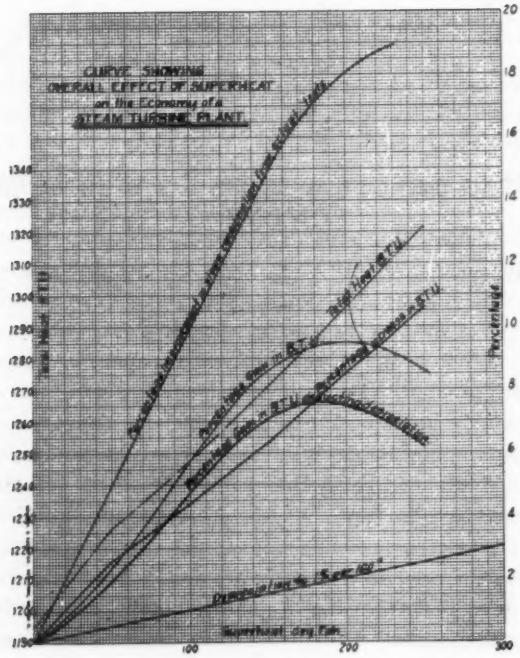
The *Delaware* is fortunate in having an excellent crew of men who like their ship, including men skilled in nearly every trade.

DISCUSSION.

AMERICAN PRACTICE IN USING SUPERHEATED STEAM.

CHIEF ENGINEER W. J. A. LONDON, OF THE TERRY STEAM TURBINE CO.

With reference to Captain Carr's paper in regard to American practice in using superheated steam, the writer is particularly interested in the conclusions arrived at to the effect that the highest overall efficiency in steam turbine work is obtained somewhere in the neighborhood of 150 degrees. Some years ago the writer made an investigation on this same question as to the probable advantages of using superheated steam to the extent that is now employed in Europe, and plotted down a curve which I enclose herewith; first showing the percentage gained in steam consumption per degree of superheat. This was taken from a Parsons turbine. The total heat in B.T.U. was then plotted as taken from Knoublaugh & Jakob's tables. I then obtained a point of maximum efficiency at 200 degrees. It appeared to me obvious that there should be more depreciation in a superheated steam plant than when using saturated steam, but how much this depreciation was is, of course, a very questionable factor. After talking with several engineers and authorities on the subject, we decided that the smallest figure we could use was 1 per cent. per 100 degrees. Again deducting this from the overall efficiency curve we get a point of maximum efficiency around about 180 degrees. If, however, this depreciation is higher than 1 per cent, this will bring this point lower. Also the fact must be taken into account that it is practicable to obtain a certain amount of superheat with practically no additional



coal consumption, so taking it all around, the curve as plotted would seem to confirm the general opinion that 150 degrees is about as high as it is commercially economical to go.

DISCUSSION.

THE MARINE-TYPE LE BLANC AIR PUMP.

By Carl George de Laval, Associate.

The interesting article under the above title contained in the Journal, Vol. XXIII, No. 1, of February, 1911, by Lieut. W. W. Smith, U. S. Navy, sets forth opinions relative to the economy resulting from high vacuum and gives data concern-

ing three installations, two of which are actual performances on the *Chester* and *Salem*, of vacuum machinery which was built six years ago. The third is an assumed performance of the Le Blanc air pump, such as the one building for the U. S. S. *Neptune*, a collier not yet in commission.

Such conclusions cannot be considered reliable when drawn from performances, two actual and one assumed, the latter based upon shore practice of the type of vacuum pump referred to and known as the Le Blanc and of which shore practice has really no direct bearing upon the results that may be obtained from such an equipment if installed on board ship. The details as given under the Le Blanc are so minutely figured out that it would appear to the casual observer that they are from actual performances, and they also appear to be put forth with the idea of drawing attention to the assumed superiority of the Le Blanc type of air pump in comparison with the others. For this purpose it is not out of the way to call attention to the various irregularities in the comparison of the steam economy and weight of the Le Blanc air pump with augmenter and twin pumps on the *Chester* and the dry-vacuum pumps on the *Salem*.

The writer hopes some day to be able to show and describe a superior equipment to them all, which may be in use in the near future for marine service.

In the tables as given by the author, the basis of assumption of the percentage of ideal vacuum as mentioned in Table 1, lines 1 to 7, is incorrect. It is based only on the discharge temperature of the circulating water and the vacuum measured only at the air pump without regard to the temperature of the air suction from the condenser, of the cold water entering the Le Blanc pump, or of the jacket water supplied to the rotative dry-vacuum pump, all of which control and limit the vacuum.

While it is not possible to obtain at the exhaust inlet of a surface condenser a vacuum in excess of that corresponding to the temperature of the discharge of circulating water, a higher vacuum may be obtained at the air pump, and to obtain such a higher vacuum at the pump it is necessary only to re-

strict the air-pump suction line, making it smaller, or obstructing same by sharp turns in valves or in such an air-suction check valve as is shown on page 51.

This may readily be shown by assuming an increased frictional resistance in the air-pump suction line. Assuming in the Le Blanc test that the discharge circulating temperature is 80 degrees as per Table No. 1, line No. 1, this temperature corresponding to an ideal vacuum of 28.98 inches as per line No. 5, we will assume such increased friction in the air-pump suction line such that Le Blanc pump, supplied with salt water at 65 degrees and with a temperature difference of 5 degrees, as per Table No. 1, line No. 5, would maintain within itself a vacuum corresponding to 70 degrees or 29.27 inches. Comparing this with the ideal vacuum of 28.98 inches, as per line No. 5, the pump would then be figured as maintaining 102 per cent. of the ideal vacuum.

Evidently to obtain a correct idea of the work done by the several pumps other data must be taken into consideration. Large, free, open-suction pipes must be used, or losses given, and the vacuum for comparison to the ideal as determined by the discharge circulating water and other data must be measured by mercury column at exhaust inlet to the condenser, and not taken at the air pump, as recorded by Lieutenant Smith.

Had such an obstruction as mentioned been placed in the air-pump suction line or were same restricted, and thus a higher vacuum maintained at the pump than in the condenser, Lieutenant Smith's method of figuring would show instead of 720 pounds of steam chargeable to the Le Blanc air pump in Table No. 1, line 10, a minus quantity, or the pump would figure out as actually supplying steam to the boilers instead of consuming it.

The ideal vacuum maintainable by an air pump of the twin-beam type as used on the *Chester* depends upon the temperature of the air, vapor and water of condensation reaching it. In a rotative dry-vacuum pump such as used on the *Salem* it depends upon the temperature of the air suction and the temperature of the jacket water surrounding the air cylinder. In

the Le Blanc it depends on the temperature of the sea suction supplied to the pump. None of these temperatures are given by Lieutenant Smith. The dry-vacuum pump on the *Salem* is apparently working under the hardest conditions in these comparative readings, as we may judge from the circulating-water discharge temperature being 7 degrees higher than that of the Le Blanc unit, that the circulating suction was also higher, and therefore the degree of temperature to which the air cooler in surface condenser could reduce the temperature of the air suction to this pump was also limited. The forementioned conditions, wherein a percentage of 102 per cent. of the ideal may be figured, conclusively proves that the percentage of ideal vacuum cannot be measured by a simple relation of vacuum reading at the air pump and temperature of discharge water, but must take into consideration the other data previously mentioned.

Lieutenant Smith does not state whether vacuum was measured by spring gage or mercury column. Such readings in competitive tests should always be measured by a full-length mercury-column gage.

The temperatures, percentages, etc., as covered in table No. 1 are figured very finely, as though from an actual test, although Lieutenant Smith does not mention the name of the vessel in which the tested Le Blanc pump is installed.

This Table No. 1 shows the steam used by the air pump per thousand pounds of steam consumed by the turbine on line No. 13 as follows: Augmenter-type pump, 27.6; dry-vacuum pump, 16.75; Le Blanc type, 25. In line No. 14 the total steam chargeable to the air pump figured upon a steam-consumption increase of 4 per cent. per inch loss in vacuum below ideal, as figured from comparison of vacuum maintained at air pump to temperature of discharge water, but for which, as previously mentioned, no supporting data is given to warrant the charging of a larger increased steam consumption for the *Chester* and *Salem* compared to the Le Blanc units. Nevertheless, these figures still show a total quantity of steam chargeable to the air pump as 42.5 pounds for the augmenter-type installation

on the *Chester,* 25.6 for the dry-vacuum pump unit upon the *Salem,* and 32.2 pounds for the Le Blanc unit. This gives a relative steam consumption as shown upon line No. 15 of .795 pound for the *Salem,* 1.32 for the *Chester,* and 1 for the Le Blanc. These results show the steam consumption of the dry-vacuum pump together with the steam chargeable to same by Lieutenant Smith's method of figuring as not quite 80 per cent. of the steam consumption of a similarly installed Le Blanc unit. Having thus failed to make a case for the Le Blanc unit, Lieutenant Smith assumes in Table 2 that a Le Blanc pump of two-thirds the capacity and steam consumption of that in Table No. 1 will do exactly the same work, thus giving 1.21 pounds for the *Salem,* 1.46 pounds for the augmenter unit upon the *Chester,* to 1 pound for the Le Blanc, but as his conclusion is apparently not based on facts derived from actual test, his figures cannot be accepted as indicative of what may be expected under such conditions.

The comparison of weights indicated in Table No. 1, lines 16, 17 and 18, is not a fair comparison when the total installation is considered, as it omits from the calculation the extra weight of the water pipes required for the Le Blanc unit and also the weight of the water contained in the Le Blanc pump and in the connecting piping.

LIEUTENANT SMITH'S REPLY.

Referring to Table I, second and fourth lines, the "air pump vacuum" referred to is the vacuum maintained at the inlet of the condenser. Being the actual vacuum maintained by the air pump, it was so called to distinguish it from the ideal vacuum corresponding to the temperature of the discharge water. While the "casual observer" might understand that this reading was taken at the suction of the air pump, it is inconceivable that one who has carefully read the text and studied the tables could fail to grasp the correct and obvious meaning. In view of this erroneous assumption in the discussion, it is not necessary to point out the absurdity

of the assumed condition where the air pump maintains 102 per cent. of the ideal vacuum and supplies steam to the boilers.

Referring to paragraph twelve of the discussion, a careful reading of the text on page 59 in connection with Table I will show why the "increased steam consumptions" (items 10) for the *Chester* and *Salem* are so much larger than that for the Leblanc. It will be seen that item 10 = item 9 × item 6 × item 8. As items 8 for the *Chester* and *Salem* are over 2½ times that for the Leblanc, it is apparent that items 10 for the *Chester* and *Salem* will naturally be much larger than that for the Leblanc. It is evident that these values are not comparable, and it was not intended that they should be. To compare the steam consumptions it is necessary to reduce them to unit values, and where comparison was made (items 13 and 14) this was done. To make this clear, suppose item 6, the difference between the ideal and actual vacuum, had been 1 inch for all three outfits. Then item 10 would have been, for the *Chester* 11,300; for the *Salem* 10,200; and for the Leblanc only 4,000. Still, when reduced to unit values, it is clear that items 10 become the same in each case or 40—*i.e.*, item 10 ÷ item 8 × 1,000. Item 11 was obtained from test data, as explained; item 12 = item 10 + item 11; item 13 = item 11 ÷ item 8 × 1,000; and item 14 = item 12 ÷ item 8 × 1,000. From the above it is obvious that the increased steam consumptions (items 10) and the steam chargeable to the air pumps (items 14) are correct, and that the assertion that "no supporting data is given to warrant the charging of a larger increased steam consumption for the *Chester* and *Salem* compared to the Leblanc units" is incorrect.

The additional weights of piping, foundations, etc., should be considered, of course. For like installations, weights of piping and foundations for the Augmenter and Leblanc types will probably be about the same; for the dry-air type, similar to the one installed on the *Salem*, these weights will probably be considerably more, because of the additional piping and foundations required for two separate units—one being of

considerable size. However, I do not believe these additional weights will materially change the relative values of the weights given.

The size or capacity of the air pump depends on the quantity of air leakage it is required to withdraw, and if the latter is known the former is determined. As reliable data were not available, the probable quantity of air leakage for a naval turbine installation of a given power could not be determined with sufficient accuracy; and, to be certain of not underestimating, it was deemed advisable to assume a value well in excess of the probable, which naturally gave the proportionately larger pump given in Table I. The quantity of air leakage will, of course, depend on the installation and its condition. The construction and service condition of an average naval turbine installation is such that the air leakage is probably about the same as that of the best class of shore installations fitted with surface condensers. In view of the above, and considering, as stated in the text, that it is the standard practice of a European firm, which has had considerable experience and success with the Leblanc pump, to use a pump one-third smaller than the one given in Table I, it is not unreasonable to expect that the smaller pump will have ample capacity; also, it is reasonable to assume that a large size will not be used when a smaller will suffice; and that the small size approximates the standard practice of the future. As the probable values for future practice are of even more importance than these for the particular unit given in Table I, it was considered desirable to present the data for the smaller size as given in Table II. It was not asserted that the smaller pump will do exactly the same work as the larger—for this is an absurdity. It should be apparent that the air leakage for the smaller pump must necessarily be smaller in proportion to the reduction of capacity.

Surely the results obtained in land installations of the Leblanc pump have a close relationship to its performance on board ship, for the construction is almost identical. We may

say that the air pump does not know whether it is afloat or ashore, provided the conditions are the same.

Mercury columns were used to measure the vacuum for the Leblanc tests, on the *Salem*, and, I believe, on the *Chester*.

The air-pump suction (piping and valves) is not restricted, as the reader may infer. The design provides for a short direct connection of ample area, so that the difference in vacuum between the condenser and air pump is necessarily small—less than 1/20 inch of mercury, estimated. Supposing its area were too small, it would be a very simple matter to provide a larger connection of ample area. It may be interesting to know that when the air suction is closed, and therefore when no air is being pumped, the pump will maintain in its air suction the ideal vacuum corresponding to the temperature of the water passing through it. (This is one of the test requirements which every pump must meet.) If the temperature of the water were 65 degrees, as taken in the discussion, the vacuum in the air pump would be the ideal or 29.377 (referred to 30-inch barometer).

Complete data for the outfits on the *Chester* and *Salem* were not available, the data being obtained from general trials, and not from special tests of the condensing plants. Attention is again invited to the text, where it is plainly stated that the data were insufficient, and the comparisons only approximate. It should therefore be clear to the reader that it was not intended otherwise, as implied in the discussion. It would indeed be interesting and instructive to have complete data of tests of the three types, made under the same conditions so that accurate comparison could be made; and it is hoped that such tests will be made and published.

While the performance of a single unit may give characteristic results, still there may be some eccentricity of the unit or test that will cause the results to be better or worse than the characteristic or average results obtained with a number of similar units; and it is therefore generally admitted that the average performance of a number of similar units is more reliable as indicative of the characteristic per-

formance than that of one particular unit. The data for the Leblanc pump were obtained from a number of tests made solely to determine the performances of the condensing plants, and particularly the air pumps. Having observed many tests on these units, I have no doubt as to the accuracy and reliability of the data. To be reasonably safe in predicting a good average performance for marine units, the assumptions were conservative; in fact, most of the test and service performances are better than the ones assumed. It is therefore reasonable to believe that the performances assumed are fairly close and conservative approximations to the results that will be obtained with marine units. The reader may infer from the discussion that the performance data given for the Leblanc pump were not based upon actual performances. This is entirely incorrect, as is apparent from the above.

In making comparisons of this nature, where complete data are not available, some values have to be assumed, and, obviously, they are not minutely accurate and absolutely reliable; but it is believed that the results given are not far in error, and that they are sufficiently accurate for the approximate comparisons made. As made clear in the text, it was not expected that the results would be accepted as absolutely correct and conclusive. It was considered instructive and desirable to set forth the relative qualities of the three types in a general way only.

NOTES.

AUSTRALIAN NAVY.

It is announced in connection with the Australian Navy that a Naval Board has been established at Melbourne, of which the Minister of Defence will be president and Captain Cresswell, Director of the Commonwealth Naval Forces, promoted to be Rear Admiral, will be senior member.

Admiral Sir Reginald Henderson, who has been inspecting the harbor and other defences of Australia, has presented his report. The fleet, which is to be completed within twenty-two years, will cost in all £23,000,000, with a personnel of 15,000 men. Sir Reginald Henderson recommends that naval construction shall be divided into quinquennial eras.

The fleet will consist of eight armored and ten protected cruisers, eighteen destroyers, and twelve submarines.

Admiral Henderson recommends the Commonwealth, either by the establishment of Government workshops or by the encouragement of private enterprize, to build up the plant and power necessary to manufacture ammunition, ordnance and shipbuilding materials, progressing gradually until all requisites, including the largest guns, can be manufactured in Australia.—"Page's Weekly."

BRITISH NAVY ESTIMATES.

The Navy Estimates for 1911–12 show a total of £44,392,500. Provision is made for an increase of 3,000 men, raising the total personnel to 134,000. New construction will include five *Dreadnoughts,* four smaller cruisers, twenty destroyers, and six submarines.

New construction will cost £15,063,877, as against £13,-

279,830 for 1910–11. Of this amount £13,325,232 will be spent on the continuation of work on the ships already under construction, and £1,738,645 for beginning work on ships of the new program, which is composed as follows: five large armored ships, three protected cruisers, one unarmored cruiser, twenty destroyers, six submarines, two river gunboats, a depot ship for destroyers, and a hospital ship. Between April 1st, 1910, and March 31st, 1911, the following ships will have been completed and become available for service: One battleship, *Neptune;* one armored cruiser, *Indefatigable;* five protected cruisers, *Bristol, Glasgow, Gloucester, Liverpool* and *Newcastle;* one unarmored cruiser, the *Blanche;* three submarines, *C33, C34* and *D2*; and the following twenty-eight torpedo-boat destroyers: *Viking, Beagle, Bulldog, Foxhound, Grasshopper, Mosquito, Scorpion, Renard, Wolverine, Racoon, Harpy, Basilisk, Savage, Pincher, Rattlesnake, Scourge, Acorn, Chameleon, Lyra, Redpole, Comet, Larne, Goldfinch, Alarm, Rifleman, Hope, Nemesis* and *Martin.*

ADMIRALTY NOTES ON WORK DONE.

Ordnance, Fleet-Coaling Service, New Construction, Machinery and Boilers, etc.

The following abstract is from the record of work done, appended by the First Lord of the Admiralty to his statement explanatory of the Navy Estimates:

ORDNANCE.

Guns.—The manufacture of guns is proceeding at a sufficient rate to meet the requirements for ships building.

Ammunition.—The authorized reserves of ammunition have been fully maintained.

Advance continues to be made in designs of all natures of projectiles.

Cordite.—Certain difficulties which arose in connection with the supply of cordite early in the year have been overcome.

Small Arms.—Short rifles have been supplied to the Royal Marines on shore and to Royal Marines afloat in the Home, Atlantic and Mediterranean Fleets. It is expected that complete supply to the Royal Marines afloat will be made during the financial year 1911–12.

The older marks of short rifles are being brought up to date as far as possible to include the improvements embodied in the latest pattern rifle.

Gun Machinery.—The manufacture of the 13.5-inch gun machinery for the *Orion* and *Lion* classes has proceeded satisfactorily during the past year, and it is anticipated that the turrets for *Orion* and *Lion* will be in position in the ships before the end of the present financial year. The trials of the first mountings have taken place with satisfactory results.

Hand-worked Mountings.—The mechanism of hand-worked mountings has been progressively developed towards obtaining increased rate of fire and economy of labor.

Gunnery Practices.—The rules for the annual tests of gunlayers were revised at the beginning of 1910, and have proved to be an advance on those previously in force. The rules for 1911 have been amended in some particulars, the same general principles being retained.

Torpedoes.—Satisfactory progress has been made in the supply to the fleet of the improved torpedo referred to last year.

Wireless Telegraphy.—Several small improvements have been made in the wireless apparatus of ships, which has led to increased efficiency.

Additional shore stations have been commenced and should be completed by next autumn.

Inspection and Proof.—The inspection of ordnance stores carried out at contractors' works is proceeding in accordance with arrangements which have been made, and considerable advantages are anticipated on the completion of the testing house, which is in course of erection at Sheffield.

Ordnance and Torpedo Depots.—The torpedo factory at Greenock has been opened and the work of manufacture has

begun. The factory at Woolwich will be closed on the 31st March, 1911.

FLEET-COALING SERVICE.

Progress continues to be made in improving and adding to the facilities for coaling the fleet with despatch and economy.

Considerable progress has been made in connection with the coaling scheme on shore at Devonport, and the machinery should be installed and in full working order during the course of 1911.

Two large hulks, each of about 10,000 tons, are also being equipped with modern coal-handling appliances, and will be available for service early in the year.

The colliers specially engaged by the Admiralty on continuous charter have proved very suitable for fleet work, and some excellent coaling performances have been made by H.M. ships in coaling from them.

Oil Fuel and Petrol.—Additions have been and are being made to the Admiralty tank installations for the storage of oil fuel and petrol to meet the growing expenditure of these fuels in the Navy.

The new fleet auxiliary (S.S. *Burma*) for the conveyance of oil fuel to the fleet is expected to be ready for service early in 1911–12.

The new petrol carrier Royal Fleet Auxiliary *Isla* has been employed in attendance on the submarine flotillas throughout the year, and has rendered valuable service.

NEW CONSTRUCTION.

The battleship *Orion*, which was laid down at Portsmouth on 29th November, 1909, was launched on the 20th August, 1910, and the armored cruiser *Lion*, laid down at Devonport on 29th November, 1909, was launched on 6th August, 1910. Substantial progress has been made during the current year on both ships, and it is expected that they will be completed within a period of two years from the date of laying down. Good progress is also being made with the four contingent

ships of the 1909–10 program, the *Conqueror, Monarch, Thunderer* and *Princess Royal.*

The protected cruisers *Falmouth, Weymouth* and *Dartmouth* were launched on 20th September, 1910, 18th November, 1910, and 14th February, 1911, respectively, and good progress is being made on them, as well as on the sister ship *Yarmouth.*

The unarmored cruiser *Blanche,* laid down at Pembroke on the 12th April, 1909, has satisfactorily carried out all her steam trials, and is now in commission. The sister ship *Blonde,* laid down on the 6th December, 1909, and launched on the 22d July, 1910, is now at the steam-trial stage.

Several of the twenty destroyers of the *Acorn* class, ordered in 1909–10, have been delivered.

Four submarine boats of the " D" class, Nos. 3, 4, 5 and 6, being built by contract, are well advanced.

Submarine boats *C*33 and *C*34, which were built and engined by Chatham Yard, were completed on the 13th August, 1910, and 17th September, 1910, respectively, and have joined the submarine flotilla. The two submarine boats of " D" type under construction at Chatham will, it is anticipated, be completed in the early months of 1911–12.

The following fighting vessels were provided for in the program for the present financial year, and orders have been placed for the whole of them.

Battleships (4).

King Georve V. — *Centurion.*
Ajax. — *Audacious.*

Armored Cruiser (1).

(Name not yet given.)

Protected Cruisers (3).

Chatham. — *Dublin.*
Southampton.

Unarmored Cruisers (2).

Active.	*Amphion.*

Torpedo-Boat Destroyers (23).

Acheron.	*Goshawk.*
Archer.	*Hind.*
Ariel.	*Hornet.*
Attack.	*Hydra.*
Badger.	*Jackal.*
Beaver.	*Lapwing.*
Defender.	*Lizard.*
Druid.	*Phoenix.*
Ferret.	*Sandfly.*
Forester.	*Tigress.*

And three others not yet named.

Submarines (6).

Of the foregoing vessels, the battleships *King George V* and *Centurion* were laid down at Portsmouth and Devonport, respectively, on January 16th, 1911, the *Ajax* was laid down at Messrs. Scott's work at Greenock on February 27th, 1911, and preparations for building the sister ship *Audacious* are well advanced at Messrs. Cammell, Laird's yard at Birkenhead. The order for the armored cruiser has been placed provisionally with Messrs. Palmer's Shipbuilding Company.

The two armored cruisers, *Australia* and *New Zealand*, for the Governments of the Commonwealth of Australia and New Zealand, are well advanced at the works of Messrs. John Brown and the Fairfield Company, respectively, and are to be completed in September, 1912.

The protected cruiser *Chatham* was laid down at Chatham Yard on January 3d, 1911, and her sister vessels *Dublin* and *Southampton* have been ordered from Messrs. Beardmore and Messrs. John Brown, respectively. Two more ships of this class have been ordered from Messrs. Cammell, Laird and

the London and Glasgow Company for the Commonwealth of Australia, and named *Melbourne* and *Sydney*, respectively. Drawings and specifications have been supplied to the Canadian Government for use in calling for tenders for four vessels of this class.

The *Active*, unarmored cruiser of the improved *Boadicea* type, which was laid down at Pembroke in July, 1910, launched on March 14th, will be followed on the same slip by the sister ship *Amphion*.

The twenty vessels of the new *Acorn* class have been ordered by contract, and good progress is being made with them. The three additional vessels included in the 1910–11 estimates have recently been ordered.

Two submarines were laid down at Chatham Dockyard on February 14th, and an order for four more has been placed with Messrs. Vickers. Two similar boats have also been provisionally ordered for the Australian Commonwealth.

Fleet Auxiliaries, etc.—A depot ship for submarines has been ordered from Messrs. Scott, of Greenock, and named *Maidstone;* and two tenders from Messrs. Cammell, Laird named *Alecto* and *Adamant*.

The oil-tank steamer *Burma*, now being built by the Greenock and Grangemouth Dock Company, is well advanced.

A coastguard and fishery cruiser, to be named *Watchful*, has been ordered from Messrs. Hall, Russell & Co., of Aberdeen.

Tenders are under consideration for a surveying vessel to replace *Egeria*.

A steel water-tank and salvage vessel for submarines has been ordered from Messrs. Vickers for delivery during the financial year 1911–12.

Orders for two floating docks have been placed provisionally with Messrs. Swan, Hunter & Co.

Tugs, Lighters, etc.—The following tugs and other small vessels have been completed:

Hellespont—paddle tug for Haulbowline.

Firm—paddle tug for Sheerness.

Alliance—twin-screw tug for Sheerness.

Resource—mooring steamer for Portsmouth.

Supply and *Urgent*—two water-tank vessels for Plymouth and Sheerness, respectively.

Two victualling lighters for Plymouth and a mooring and salvage lighter for Sheerness.

One ammunition and store barge.

One lighter for Ascension. One steam ash-hopper barge for Dover.

One mooring lighter for Sheerness is in hand.

Vessels Being Converted for Various Services.—Naiad and *Intrepid* have been completed for service as mine layers.

Rosario is being converted into a submarine depot ship.

MACHINERY AND BOILERS.

All war vessels completed during the present year and at present under construction have been designed with water-tube boilers and turbine engines, the latter being either of the Parsons or of the Brown-Curtis type. This latter design was first intsalied in the second-class cruisers *Bristol* in association with the superheaters at the boilers, and satisfactory results were obtained on the contractors' trials.

So far as can be arranged, interchangeability of the auxiliary machinery and separate parts of the main machinery is being provided for in vessels of the same type, thus following previous practice.

Additional machines have been added to the equipment of *Assistance* and *Cyclops,* enabling them still further to deal with repairs which otherwise would have required dockyard assistance.

Magazine cooling plant has been fitted in all vessels completed or at present under construction.

The boilers of all armored vessels building are being fitted to burn oil in conjunction with coal, the full power being obtainable in these vessels by the use of coal only. Satisfactory full-power trials have been carried out in the second-class

cruisers of the *Bristol* class and *Blanche*, the boilers of which vessels are designed to burn oil in conjunction with coal when obtaining full power.

The torpedo-boat destroyers of this year's program are designed to burn oil only, and improvements have been made in the oil-burning fittings.

Experiments have been made and are still in progress at Haslar to reduce the excessive amount of smoke produced when burning bituminous coal in the furnaces of water-tube boilers; a substantial reduction has already been effected.

Internal-Combustion Engines.—Valuable experience with the oil-driven dynamo engines fitted in H.M. ships has been obtained, and the results are being closely watched.

The initial difficulties experienced with the engines of motor boats using heavy oil are being gradually overcome.

Small petrol or paraffin-driven motor boats have been supplied to two vessels of the destroyer class and have proved very useful. All new torpedo-boat destroyers will be provided with similar boats.—"Page's Weekly."

WARRANT RANK IN THE BRITISH NAVY.

In the course of his observations on the personnel of the Navy the First Lord remarks that, as the seniority of the artificer engineers at the head of the list rises, steps will be taken to increase the number of chief artificer engineers within the limits already authorized. During the last financial year ten artificer engineers were advanced to chief rank, an increase of four over the preceding year. As foreshadowed in 1905, the new ranks of warrant mechanician (warrant officer) and commissioned mechanician (commissioned warrant officer) have been instituted, and five mechanicians have been promoted to warrant rank. Candidates for the new warrant rank sat for the same examination as that taken by candidates for the rank of artificer engineer, and the

result was very satisfactory. The number of warrant mechanicians will be gradually increased until a sufficient number has been obtained, and promotions to commissioned mechanician will be made in due course. In continuation of the policy of extending the opportunity of obtaining warrant rank beyond those classes of naval ratings to which hitherto this privilege has been restricted, warrant rank has now been introduced for the armorer and electrician branches, and the rank of commissioned warrant officer has also been established for the latter. The opportunity has also been taken to assimilate the service qualification for the advancement of electricians to the rating of chief electrician to that in force for engine-room artificers.

OIL ENGINE PROPULSION.

A great deal has been written of late about vessels, both large and small, driven by Diesel oil engines. As a matter of fact, the number of oil engines used to propel seagoing vessels probably does not as yet number more than two or three, and by far the largest of these is the *Toiler,* designed and built by Messrs. Swan, Hunter, Wigham Richardson, Ltd., at their Neptune Works, and which has recently completed a voyage from the Tyne to Calais and back. About twelve months ago, after careful investigation and consideration of various types of engines, oil, gas, electric, and others, Messrs. Swan, Hunter & Wigham Richardson, Ltd., decided to build a vessel suitable for special service on the Canadian Canals and Lakes, and to fit her with two sets of Diesel oil engines. The speed on the Canals is limited, so that engines of relatively small power were sufficient to meet the requirements. The *Toiler* is a twin-screw vessel, 248 feet long, by 42½ feet beam by 19 feet depth molded, propelled by two sets of two-cycle reversible Diesel engines of a combined B.H.P. of 360, equal to about 400 I.H.P. working at about 250 revolutions. These engines were supplied by the Diesel Motor Co., of

Stockholm, and fitted on board by Messrs. Swan, Hunter & Wigham Richardson, Ltd., at their Neptune Works. On her voyage to Calais the *Toiler* was loaded with 2,650 tons deadweight of coal cargo, besides about 40 tons of oil fuel, fresh water and stores, a total deadweight of nearly 2,700 tons on a mean draught of 14 feet. She left the Tyne in very rough weather, notwithstanding which the engines worked perfectly satisfactorily, and completed her voyage to Calais at an average speed of 5.9 knots (or 6¾ miles). On her return voyage, light in ballast, the average speed was 8.2 knots (or 9½ miles). The consumption of oil fuel for the round voyage, including auxiliary compressor, was 6½ tons, say, 1.65 to 1.75 tons per day.

FUEL ECONOMY.

In many ways the *Toiler* is more economical than a steam-driven boat. She has a greater deadweight capacity, owing to the fact that the Diesel engines are much lighter than steam engines and boilers, the amount in this vessel being about 60 tons; the cubic capacity for cargo is also greater, as the boiler space is saved, and besides the oil fuel can be carried in the double bottom in place of water ballast, thus saving bunker space. The consumption of oil is much less in weight than the consumption of coal for steam engines. In the *Toiler* the consumption at full speed does not exceed 1.75 tons per day of fuel oil of 18,000 B.T.U. calorific value, whereas with steam engines of equal power the consumption of coal would not be less than 8 tons per day. The actual difference in cost depends, of course, upon the relative prices of coal and oil in the district where the vessel may be trading. Not only is the economy of this vessel shown in the increased deadweight of cargo carried, the increased cubic capacity and the low fuel consumption, but the staff on board to attend to the Diesel engines is less than the staff required for steam engines and boilers, the engine-room staff being about the same, whilst firemen are not required. The deck machinery and en-

gine-room accessories of the *Toiler* are driven by compressed air being furnished by a compressor driven by a small Diesel engine. The electric light, with which the vessel is fitted, obtains its power from a small paraffin engine. The accommodation is heated by hot water, and the heat is obtained either from the exhaust gases of the main engine or by means of a coal fire. There is thus no steam on board the boat. Messrs. Swan, Hunter & Wigham Richardson, Ltd., have now taken up the construction of marine Diesel oil engines in their own Neptune Engine Works, and have at present a set of four-cylinder two-cycle engines in hand.—"Page's Weekly."

SHIPS.

DENMARK.

The first-class torpedo boat *Soridderen,* built for the Danish Government by Messrs. Yarrow, of Glasgow, was launched on Tuesday, February 28th.

The vessel is 181 feet 9 inches long by 18 feet beam, and is to have a speed of 27 knots. She is fitted with Brown-Curtis turbines of about 4,000 shaft horsepower, constructed by Messrs. Yarrow, steam being supplied by two Yarrow water-tube boilers of the latest type.

The *Soridderen* is to serve as a type for other torpedo boats which are being constructed in Denmark.

FRANCE.

Condorcet.—The speed trials of the French battleship *Condorcet* were carried out on January 13th to 20th. This vessel, built by the "Ateliers et Chantiers de la Loire" at Saint Nazaire, has a displacement of 18,000 tons, and a shaft-horsepower of 22,500. She is fitted with twenty-six Niclausse boilers of the latest improved type, made by Messrs. J. & A Niclausse, of Paris, and eight turbines, built by the "Chantiers de l'Atlantique," of Saint Nazaire. She is the first French *Dreadnought* to carry out speed trials. The full-power trial, as provided for in the contract, consisted of a run of ten hours with all boilers in use, and burning 26½ pounds of coal per square foot of grate surface per hour. The result of the trial gave an average speed under the above conditions of 19.257 knots as against 19.25 knots provided for. The coal consumption per knot on this trial was 2,101 pounds. The Naval Ministry further asked Messrs. J. & A. Niclausse if

they would consent to a full-power trial with a combustion of 33.34 pounds of coal per square foot of grate, which was accepted. The trial under these conditions was carried out on the 16th January, and the average speed attained over a ten-hour run was 19.80 knots, more than half a knot over the contract speed, and with a coal consumption per knot of 2,628 pounds. The *Condorcet* further carried out her endurance trials on January 20th and 21st. This trial consisted of a run of twenty-four hours with normal combustion, and the average speed attained was 18.2 knots, with a coal consumption per mile of 1,481 pounds, which was 26 pounds lower than was allowed in the contract. The performance of the turbines and boilers was perfectly satisfactory throughout all the trials; the desired steam pressure being easily maintained under all conditions.

GERMANY.

Launch of the Kaiser.—The turbine battleship *Kaiser*, launched last week, commences Germany's second squadron of eight *Dreadnoughts*. Five of these vessels are already in various stages of construction, and the remaining three will be laid down this spring the first instalments for them having been voted in the current estimates. From the laying of the keel the construction of the *Kaiser* has so far occupied eighteen months, and an equal period is expected to elapse before she is ready for sea. She is the first battleship to be built at the Kiel yard since the early nineties.

GREAT BRITAIN.

Launch of the Monarch.—H.M.S. *Monarch*, launched by Sir W. G. Armstrong, Whitworth & Co., at Elswick, is one of the four "contingent" vessels ordered at the close of 1909–10.

According to particulars which have appeared in the daily

press, the main armament of the new battleship consists of ten 13.5-inch guns mounted in pairs in barbettes on the middle line of the ship. All the ten guns are available on either broadside, but only four are capable of use ahead and four of use astern, as compared with an ahead fire of six guns and an astern fire of eight guns in the case of the *Neptunes*.

Her thickest armor is 12 inches, as compared with 10 inches in the cases of the three *Neptunes* and the three *St. Vincents*, and 11 inches in the cases of *Dreadnought* and the three *Bellerophons*. She has been exactly a year on the stocks. Her official laying-down date was April 1st of last year, but her construction was necessarily delayed by the lock-out of boilermakers.

The principal characteristics of H.M.S. *Monarch* may be summarized as follows: Length (b.p.), 545 feet; beam, 88½ feet; draught, 27½ feet; displacement, 22,500 tons; machinery, Parsons turbines; shaft horsepower, 27,000; boilers, Yarrow; speed, 21 knots; big guns, ten 13½ inches; broadside fire, 10 guns; ahead fire, 4 guns; astern fire, 4 guns; weight of projectile, 1,250 pounds; smaller guns, sixteen 4-inch; thickest armor, 12 inches.

OBITUARY.

FREDERICK CLAY BOWERS.

Frederick Clay Bowers, "Joe" Bowers to his service friends, was born at Newark, New Jersey, on January 21st, 1858. In September of 1875, he was appointed to the Naval Academy, after competitive examination, as a Cadet Engineer. He was duly graduated in 1879, and began his duties as a Naval Engineer. These duties extended over a period of many years and were always performed in a spirit of high zeal and loyalty and with an energy and thoroughness that made his record an enviable one. It is not necessary to name the ships he served on, nor where they carried him. His shipmates and his shoremates who read this do not have to be reminded that he performed his duties well. What they will want to know is something of his last days and how he met the summons to quit this life.

In the spring of 1910, while on duty as inspector of machinery in the vicinity of New York, he suffered a physical breakdown, something of the nature of paralysis of the lower limbs. Knowing himself to be incapacitated for further work, he requested and was granted retirement on June 30th, 1910.

The writer saw much of "Joe" during the days his illness was developing and after it reached the hopeless stage. His manly cheerfulness during these days was something to wonder at and admire. In the sudden collapse of a splendid strength he met the prospect of years of helplessness without a word of protest against his hard luck. Living pluckily to the end, his illness proved the stronger, and he finally died on April 17th of this year.

He has passed through the door that will open to all of us sooner or later. Those who care for us can be proud indeed if we make our exit as did our friend, unafraid and with smiling acceptance of what had to be.—G. W. M.

AMERICAN SOCIETY OF MECHANICAL ENGINEERS—SPRING MEETING.

The local committee of the American Society of Mechanical Engineers, E. M. Herr, Chairman, Elmer K. Hiles, Secretary, having in charge the preparations for the convention of the Society, which will be held in Pittsburgh, Pa., May 30th to June 2d, inclusive, has completed the work of arranging the program for each day during the meeting.

It has been settled that the arriving guests will be received and registered at the Hotel Schenley, the Society headquarters, on Tuesday morning, May 30th. In the evening there will be an informal reception for the members and ladies in the parlors of the hotel. The extensive alterations being made by the new management of the hotel will add largely to the comfort and enjoyment of visitors during the convention.

Professional sessions will be held in the Lecture Hall of the Carnegie Institute, near the headquarters, Wednesday morning and evening, Thursday and Friday morning. In the meantime there will be a number of inspection trips to various industrial plants in the vicinity; a boat excursion for the members and ladies up the Monongahela River, a reception and ball at the Hotel Schenley on Thursday evening and, finally, on Friday evening a smoker and entertainment, given by the Engineers' Society of Western Pennsylvania in their rooms in the Oliver Building.

A carefully prepared program for the entertainment of the lady visitors has been arranged by a committee of ladies from Pittsburgh and vicinity, which includes a number of delightful social functions.

Judging from the number of inquiries which have already been received from expected guests, a very large attendance is looked for, and everything indicates that this meeting in Pittsburgh will be one of the most successful "Spring Meetings" the Society has ever held.

ASSOCIATION NOTES.

The following Members and Associates have been received into the Society since the publication of the list of members in the February number of the JOURNAL:

MEMBERS.

Beach, E. L., Commander, U. S. Navy.
Campbell, J. A., Lieutenant, U. S. Navy.
Claude, A., Lieutenant, U. S. Navy.
Dawes, R. A., Lieutenant, U. S. Navy.
Larimer, E. B., Lieutenant, U. S. Navy.
McCracken, John J., Lieutenant, U. S. Navy.
Miller, W. S., Lieut. Commander, U. S. Navy.
Minor, Lucian, Lieutenant, U. S. Navy.
Stuart, H. A., Lieutenant, U. S. Navy.
Todd, D. W., Lieut. Commander, U. S. Navy.

ASSOCIATES.

Andrews, George M., General Manager, New York Shipbuilding Co., Camden, N. J.
Beeman, Robert, Engineer Lieutenant, R. N., c/o Admiralty, Whitehall, London, S. W., England.
Benitez, Jose M., Lieut. Commander Engineer, Argentine Navy, c/o Fore River Shipbuilding Co., Quincy, Mass.
Buckingham, Edgar, Bureau of Standards, Washington, D. C.
Chapman, Lawrence, Lake Torpedo Boat Co., Bridgeport, Conn.
de Bertodano, Juan L., Lieut. Commander Engineer, Argentine Navy, c/o New York Shipbuilding Co., Camden, N. J.
Deiser, Charles E., c/o New York Shipbuilding Co., Camden, N. J.
Eynon, George E., 225 Lees Avenue, Collingswood, N. J.

Ford, Henry R., Lieutenant and Engineer, New York N. M., 114 Windsor Avenue, Buffalo, N. Y.

Gardner, Albert H., c/o New York Shipbuilding Co., Camden, N. J.

Hansen, Paul A., c/o New York Shipbuilding Co., Camden, N. J.

Harris, Herbert Henry, c/o New York Shipbuilding Co., Camden, N. J.

Howitz, A. A., c/o New York Shipbuilding Co., Camden, N. J.

Kauffmann, Frederick F., 909 Pine Street, Camden, N. J.

Keith, H. W., Massachusetts Institute of Technology, Boston, Mass.

McKinney, John S., Bureau of Steam Engineering, Navy Department, Washington, D. C.

Olivera, Emilio M., Commander Engineer, Argentine Navy, 95 Merrymount Road, Quincy, Mass.

Pistor, Philip, 227 East Main Street, Moorestown, Burlington Co., N. J.

Pistrelli, Atilio, Lieutenant Engineer, Argentine Navy, c/o New York Shipbuilding Co., Camden, N. J.

Snow, Norman Leslie, 1342 Watchung Avenue, Plainfield, N. J.

Sommer, Albert, 30 Church Street, New York City.

Stout, R. Paul, Ordnance Engineer, Bethlehem Steel Co., Bethlehem, Pa.

Treder, H. M., Chief Engineer, N. A. S., U. S. S. *Vestal.*

Turner, Ernest G., 1305 Locust Street, Philadelphia, Pa.

Vanderbilt, Cornelius, 30 Pine Street, New York City.

Villacian, Zacaries, Lieutenant Engineer, Argentine Navy, c/o Fore River Shipbuilding Co., Quincy, Mass.

The following correction in the List of Members issued with the February number should be made:

Rigg, Ernest H., Naval Architect, New York Shipbuilding Co., Camden, N. J.

ERRATA.

Volume XXIII, No. 1, February, 1911.

Page 252, line 17, for "eighteen" read "nineteen."

Volume XXIII, No. 2, May, 1911.

Table facing page 414. Third column under *Warrington*, second item, read "618."

Same column, item 8, for "165,700," read "16,570."

Page 418, line 3. The reference to *H. M. S. Bristol* is in error, as that vessel has simple turbines.

Page 432, third line from bottom, for "V^2," read "V_2."

Chart facing page 458. Scale indicated in title is in error.

Page 467, thirteenth line from bottom, for "θ," read "θ_1."

Page 468, fourth line from bottom, for "Head" read "Read."

Page 473, twenty-first line, strike out second "of."

Page 475, top of page, lower line of formula, for "H_2" read "H_1."